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WORM GEARING

CALCULATION OF WORM GEARS – HOBS
SELF-LOCKING WORM GEARING

FOURTH REVISED AND ENLARGED EDITION

MACHINERY'S REFERENCE BOOK NO. 1
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NUMBER 1

WORM GEARING

FOURTH REVISED AND ENLARGED EDITION

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CHAPTER I

CALCULATING THE DIMENSIONS OF WORM GEARING*

The present chapter contains a compilation of rules for the calculation of the dimensions of worm gearing, expressed with as much simplicity and clearness as possible. No attempt has been made to give rules for estimating the strength or durability of worm gearing, although the question of durability, especially, is the determining factor in the design of worm gearing. If the worm and wheel are so proportioned as to have a reasonably long life under normal working conditions, it may be taken for granted that the teeth are strong enough for the load they have to bear. No simple rules have ever been proposed for proportioning worm gearing to suit the service it is

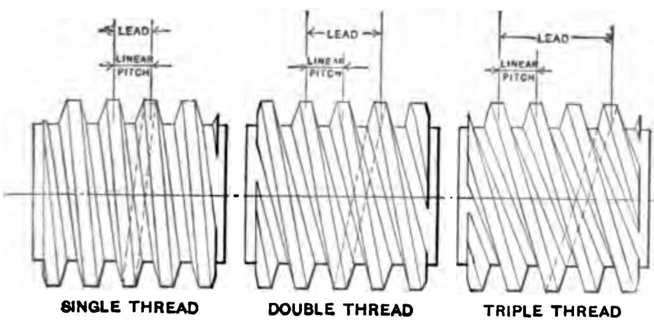

Fig. 1. Distinction between the Terms Lead and Linear Pitch as Applied to Worms

designed for. Judgment and experience are about the only factors the designer has for guidance. In Europe, a number of builders are regularly manufacturing worm drives, guaranteed for a given horse-power at a given speed. The dimensions of these drives are not made public, however; they would doubtless be of great value for purposes of comparison if they could be obtained. In the absence of these or other practical data, this phase of the subject has, of necessity, not been entered upon.

Definitions and Rules for Dimensions of the Worm

In giving names to the dimensions of the worm, there is one point in which there is sometimes confusion. This relates to the distinction between the terms "pitch" and "lead." In the following we will adhere to the nomenclature indicated in Fig. 1. Here are shown three worms, the first single-threaded, the second double-threaded, and the

* MACHINERY, August, 1907.

last triple-threaded. As shown, the word "lead" is assumed to mean the distance which a given thread advances in one revolution of the worm, while by "pitch," or more strictly, "linear pitch," we mean the distance between the centers of two adjacent threads. As may be clearly seen, the lead and linear pitch are equal for a single-threaded worm. For a double-threaded worm the lead is twice the linear pitch, and for a triple-threaded worm it is three times the linear pitch. From this we have:

RULE 1. *To find the lead of a worm, multiply the linear pitch by the number of threads.*

It is understood, of course, that by the number of threads is meant, not the number of threads per inch, but the number of threads in the whole worm—one, if it is single-threaded, four, if it is quadruple-threaded, etc. Rule 1 may be transposed to read as follows:

RULE 2. *To find the linear pitch of a worm, divide the lead by the number of threads.*

The standard form of worm thread, measured in an axial section

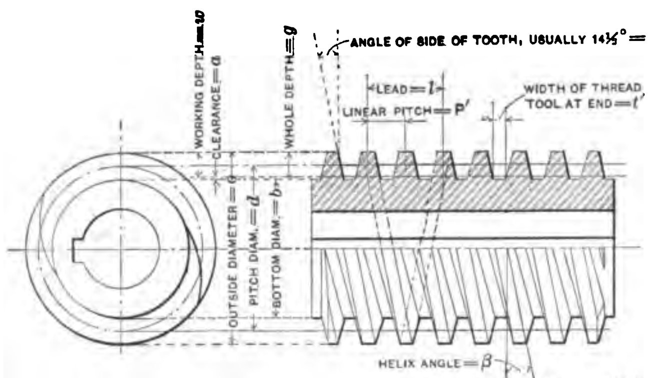

Fig. 2. Nomenclature of Worm Dimensions

as shown in Fig. 2, has the same dimensions as the standard form of involute rack tooth of the same linear or circular pitch. It is not of exactly the same shape, however, not being rounded at the top, nor provided with fillets. The thread is cut with a straight-sided tool, having a square, flat end. The sides have an inclination with each other of 29 degrees, or 14½ degrees with the center line. The following rules give the dimensions of the teeth in an axial section for various linear pitches. For nomenclature, see Fig. 2.

RULE 3. *To find the whole depth of the worm tooth, multiply the linear pitch by* 0.6866.

RULE 4. *To find the width of the thread tool at the end, multiply the linear pitch by* 0.31.

RULE 5. *To find the addendum or height of worm tooth above the pitch line, multiply the linear pitch by* 0.3183.

RULE 6. *To find the outside diameter of the worm, add together the pitch diameter and twice the addendum.*

RULE 7. *To find the pitch diameter of the worm, subtract twice the addendum from the outside diameter.*

RULE 8. *To find the bottom diameter of the worm, subtract twice the whole depth of tooth from the outside diameter.*

RULE 9. *To find the helix angle of the worm and the gashing angle of the worm-wheel tooth, multiply the pitch diameter of the worm by* 3.1416, *and divide the product by the lead; the quotient is the cotangent of the tooth angle of the worm.*

Rules for Dimensioning the Worm-Wheel

The dimensions of the worm-wheel, named in the diagram shown in Fig. 3, are derived from the number of teeth determined upon for it,

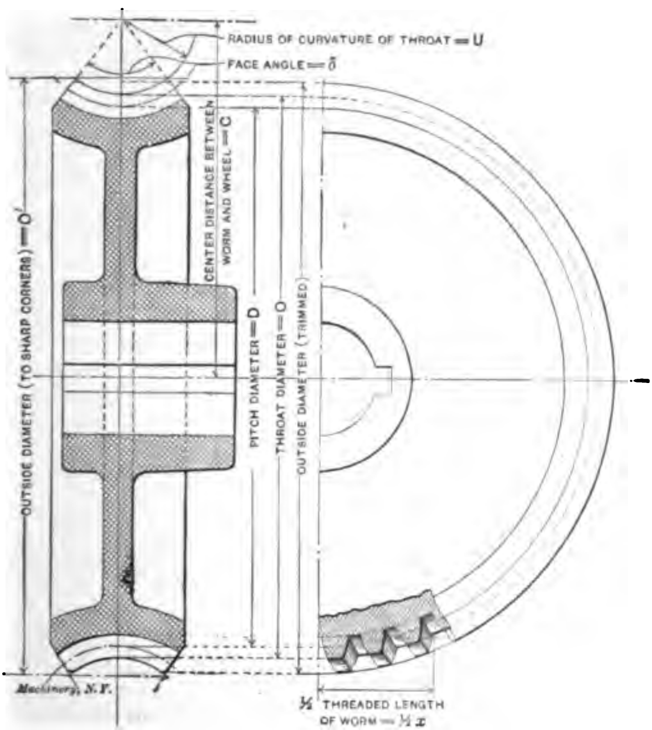

Fig. 3. Nomenclature of Worm-wheel Dimensions

and the dimensions of the worm with which it is to mesh. The following rules may be used:

RULE 10. *To find the pitch diameter of the worm-wheel, multiply the number of teeth in the wheel by the linear pitch of the worm, and divide the product by* 3.1416.

RULE 11. *To find the throat diameter of the worm-wheel, add twice the addendum of the worm tooth to the pitch diameter of the worm-wheel.*

RULE 12. *To find the radius of curvature of the worm-wheel throat, subtract twice the addendum of the worm tooth from half the outside diameter of the worm.*

The face angle of the wheel is arbitrarily selected; 60 degrees is a good angle, but it may be made as high as 80 or even 90 degrees, though there is little advantage in carrying the gear around so great a portion of the circumference of the worm, especially in steep pitches.

RULE 13. *To find the diameter of the worm-wheel to sharp corners, multiply the throat radius by the cosine of half the face angle, subtract this quantity from the throat radius, multiply the remainder by 2, and add the product to the throat diameter of the worm-wheel.*

If the sharp corners are flattened a trifle at the tops, as shown in Figs. 3 and 5, this dimension need not be figured, "trimmed diameter" being easily scaled from an accurate drawing of the gear.

There is a simple rule which, rightly understood, may be used for obtaining the velocity ratio of a pair of gears of any form, whether spur, spiral, bevel, or worm. The number of teeth of the driven gear, divided by the number of teeth of the driver, will give the velocity ratio. For worm gearing this rule takes the following form:

RULE 14. *To find the velocity ratio of a worm and worm-wheel, divide the number of teeth in the wheel by the number of threads in the worm.*

Be sure that the proper meaning is attached to the phrase "number of threads" as explained before under Rule 1. The revolutions per minute of the worm, divided by the velocity ratio, gives the revolutions per minute of the worm-wheel.

RULE 15. *To find the distance between the center of the worm-wheel and the center of the worm, add together the pitch diameter of the worm and the pitch diameter of the worm-wheel, and divide the sum by 2.*

RULE 16. *To find the pitch diameter of the worms, subtract the pitch diameter of the worm-wheel from twice the center distance.*

The worm should be long enough to allow the wheel to act on it as far as it will. The length of the worm required for this may be scaled from a carefully-made drawing, or it may be calculated by the following rule:

RULE 17. *To find the minimum length of worm for complete action with the worm-wheel, subtract four times the addendum of the worm thread from the throat diameter of the wheel, square the remainder, and subtract the result from the square of the throat diameter of the wheel. The square root of the result is the minimum length of worm advisable.*

The length of the worm should ordinarily be longer than the dimension thus found. Hobs, particularly, should be long enough for the largest wheels they are ever likely to be called upon to cut.

Departures from the Above Rules

The throat diameter of the wheel and the center distance may have to be altered in some cases from the figures given by the preceding

rules. If worm-wheels with small numbers of teeth are made to the dimensions given, it will be found that the flanks of the teeth will be partly cut away by the tops of the hob teeth, so that the full bearing area is not available. The matter becomes serious when there are less than 25 teeth in the worm-wheel. There are two ways of avoiding the difficulty. One of them is to increase the included angle of the sides of the thread tool. This departure from standard form, however, may be avoided by an increase in the throat diameter of the wheel, and consequently in the center distance. Discussions of this subject will be found in "Formulas in Gearing," and "Practical Treatise on Gearing," both published by the Brown & Sharpe Mfg. Co., Providence, R. I.

On the other hand, some designers claim to get better results in efficiency and durability by making the throat diameter of the worm-wheel *smaller* than standard, where it is possible to do so without too much under-cutting. A discussion of this subject will be found in Chapter IV of this treatise. In no case, however, should the throat diameter ever be made so small as to produce more interference than is met with in a standard 25-tooth worm-wheel.

Two Applications of Worm Gearing

Worm-wheels are used for two purposes. They may be employed to transmit power where it is desired to make use of the smoothness of action which they give, and the great reduction in velocity of which they are capable; instances of this application of worm gearing are found in the spindle drives of gear cutters and other machine tools. They are also used where a great increase in the effective power is required; in this case advantage is generally taken of the possibility of making the gearing self-locking. Such service is usually intermittent or occasional, and the matter of waste of power is not of so great importance as in the first case. Examples of this application are to be found in the adjustments of a great many machine tools, in training and elevating gearing for ordnance, etc. Calculations for the general design of this class of gearing will be treated separately. (See Chapter VI.) In the case of elevator gearing and worm feeds for machinery, the functions of the gearing are, in a measure, a combination of those in the two applications.

Examples of Worm Gearing Figured from the Rules

To show how the rules given above may be applied, we will work out two examples. The first of these is for a light machine tool spindle drive, in which power is to be transmitted continuously. It is determined that the velocity ratio shall be 8 to 1, and that the proper linear pitch to give the strength and durability required shall be about ¾ inch; the center distance is required to be 5 inches exactly. This case comes under the first of the two applications just described.

Assume, for instance, 32 teeth in the wheel, and a quadruple-thread worm. We will figure the gearing with these assumptions, and see if it appears to have practical dimensions.

The pitch diameter of the worm-wheel by Rule 10 is found to be

$$\frac{32 \times \frac{3}{4}}{3.1416} = 7.6394 \text{ inches.}$$

The pitch diameter of the worm by Rule 16 is found to be

$$(2 \times 5) - 7.6394 = 2.3606 \text{ inches.}$$

The addendum of the worm thread by Rule 5 is found to be

$$0.3183 \times \tfrac{3}{4} = 0.2387 \text{ inch.}$$

The outside diameter of the worm by Rule 6 is found to be

$$2.3606 + (2 \times 0.2387) = 2.8380 \text{ inches.}$$

For transmission gearing the angle of inclination of the worm thread should be not less than 18 degrees or thereabouts, and the nearer 30 or even 40 degrees it is, the more efficient will it be. From Rule 1 we find the lead to be $4 \times \frac{3}{4} = 3$ inches.

The helix angle of the worm thread is found from Rule 9, $2.3606 \times 3.1416 \div 3 = 2.4722 =$ cot. 22 degrees, approximately. This angle will give fairly satisfactory results. The calculations are not carried any further with this problem, whose other dimensions are determined from those just found. In the following case, however, all the calculations are made.

For a second problem let it be required to design worm feed gearing for a machine to utilize a hob already in stock. This hob is double-threaded, ½ inch linear pitch, and 2½ inches diameter. The center distance of the gearing is immaterial, but it is decided that the worm-wheel ought to have about 45 teeth to bring the ratio right. The only calculations made are those necessary for the dimensions which would appear on the shop drawing.

To find the lead, use Rule 1: $0.5 \times 2 = 1.0$ inch.

To find the whole depth of the worm tooth, use Rule 3: $0.5 \times 0.6866 = 0.3433$ inch.

To find the addendum, use Rule 5: $0.5 \times 0.3183 = 0.15915$ inch.

To find the pitch diameter of the worm, use Rule 7: $2.5 - 2 \times 0.15915 = 2.1817$ inches.

To find the bottom diameter of the worm, use Rule 8: $2.5 - 2 \times 0.3433 = 1.8134$ inch.

To find the gashing angle of the worm-wheel, use Rule 9: $2.18 \times 3.14 \div 1 = 6.845 =$ cot. 8 degrees 20 minutes, about.

To find the pitch diameter of the worm-wheel, use Rule 10: $45 \times 0.5 \div 3.1416 = 7.1620$ inches.

To find the throat diameter of the worm-wheel, use Rule 11: $7.1620 + 2 \times 0.15915 = 7.4803$ inches.

To find the radius of the throat of the worm-wheel, use Rule 12: $(2.5 \div 2) - (2 \times 0.15915) = 0.9317$ inch.

The angle of face may be arbitrarily set at, say, 75 degrees, in this case. The "trimmed diameter" is scaled from an accurate drawing, and proves to be 7.75 inches.

To find the distance between centers of the worm and wheel, use Rule 15: $(2.1817 + 7.1620) \div 2 = 4.6718$ inches.

To find the minimum length of threaded portion of the worm, use

Rule 17: $7.4803 - 4 \times 0.15915 = 6.8437$

$$\sqrt{7.4803^2 - 6.8437^2} = 3 \text{ inches, approximately.}$$

It will be noted that the ends of the threads in Fig. 2 are trimmed at an angle instead of being cut square down, as in Fig. 1. This gives a more finished look to the worm. It is easily done by applying the sides of the thread tool to the blank just before threading, or it may be done as a separate operation in preparing the blank, which will in either case have the appearance shown in Fig. 4. The small diameters at either end of the blank in Fig. 4 should, in any event, be turned exactly to the bottom diameter shown in Fig. 2, and obtained by Rule 8. This is of great assistance to the man who threads the worm,

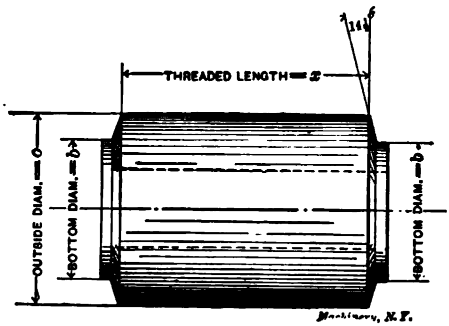

Fig. 4. Shape of Blank for Worm

as he knows that the threads are sized properly as soon as he has cut down to this diameter with the end of his thread tool. This always supposes, of course, that the thread tool is accurately made.

Formulas for the Design of Worm Gearing

For the convenience of those who prefer to have their rules compressed into formulas, they are so arranged in the following. The reference letters used are as follows:

N = number of teeth in worm-wheel.
n = number of teeth or threads in worm.
P' = circular pitch of wheel and linear pitch of worm.
l = lead of worm.
g = whole depth of worm tooth.
t' = width of the thread tool at the end.
s = addendum or height of worm tooth above pitch line.
o = outside diameter of the worm.
d = pitch diameter of the worm.
b = bottom or root diameter of the worm.
β = helix angle of worm and gashing angle of wheel.
δ = face-angle of worm-wheel.
D = pitch diameter of the worm-wheel.
O = throat diameter of the worm-wheel.
O' = diameter of the worm-wheel to sharp corners.
U = radius of curvature of the worm-wheel throat.

R = velocity ratio.
C = distance between centers.
x = threaded length of worm.

$$l = n \times P' \quad (1)$$
$$P' = l \div n \quad (2)$$
$$g = 0.6866\,P' \quad (3)$$
$$t' = 0.31\,P' \quad (4)$$
$$s = 0.3183\,P' \quad (5)$$
$$o = d + 2s \quad (6)$$
$$d = o - 2s \quad (7)$$
$$b = o - 2g \quad (8)$$

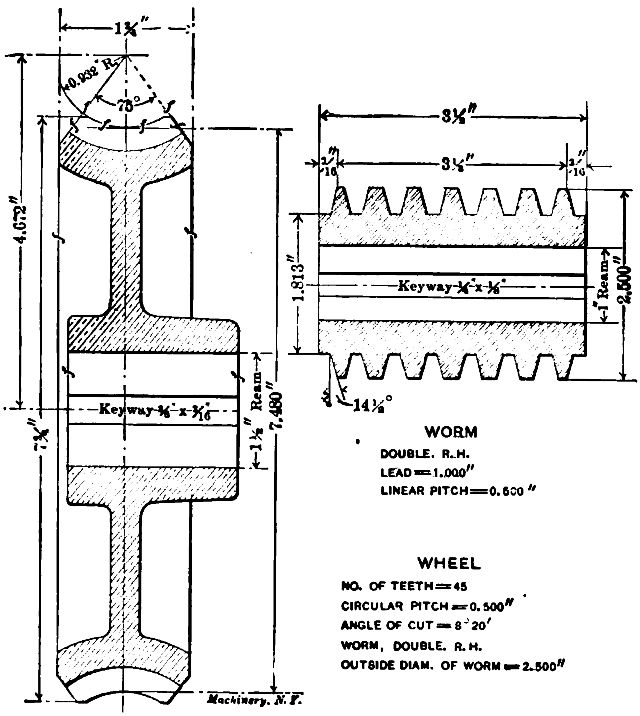

Fig. 5. Model Drawing of Worm and Worm-wheel

$$\text{Cotangent } \beta = 3.1416d \div l \quad (9)$$
$$D = N\,P' \div 3.1416 \quad (10)$$
$$O = D + 2s \quad (11)$$
$$U = \tfrac{1}{2}o - 2s \quad (12)$$
$$O' = 2\,(U - U \cos \delta/2) + O \quad (13)$$
$$R = N \div n \quad (14)$$
$$C = (D + d) \div 2 \quad (15)$$
$$d = 2C - D \quad (16)$$
$$\text{Minimum value of } x = \sqrt{O^2 - (O - 4s)^2} \quad (17)$$
$$\text{Width of worm-wheel at root of teeth} = 0.6o \text{ (approx.)} \quad (18)$$

A model drawing of a worm-wheel and worm, properly dimensioned, is shown in Fig. 5. This drawing follows, in general, the model drawings shown by Mr. Burlingame in the August, 1906, issue of MACHINERY, taken from the drafting-room practice of the Brown & Sharpe Mfg. Co. In cases where the worm-wheel is to be gashed on the milling machine before hobbing, the angle at which the cutter is set should also be given. This is the same as the angle of worm tooth found by Rule 9. In cases where the wheel is to be hobbed directly from the solid by a positively geared hobbing machine, this information is not needed. It might be added that it is impracticable with worm-wheels having less than 16 or 18 teeth to gash the wheel, and then hob it when running freely on centers, if the throat diameter has been determined by Rule 11.

When worms have a large helix angle (15 degrees or more), the dimensions of the tooth should be measured at right angles to the helix. In such cases, the following changes should be made in the formulas just given.

Let $P'_n =$ normal circular pitch $= P' \cos \beta$.

Formulas (3), (4), and (5), and the corresponding rules, should then be written as follows:

$$g = 0.6866\, P'_n \qquad (3)$$

$$t' = 0.31\, P'_n \qquad (4)$$

$$s = 0.3183\, P'_n \qquad (5)$$

When these changes are made, all the other formulas will give correct results when used in their original form.

CHAPTER II

HOBS FOR WORM-GEARS*

If a worm and gear of standard proportions are brought into mesh, we have at the bottom of both the thread of the worm and teeth of the gear a clearance equal to one-tenth of the thickness of the thread or tooth at the pitch line. The clearance at the root of the gear tooth is obtained by enlarging the hob over the diameter of the worm, by an amount equal to two clearances, while the clearance of the tooth in the thread bottom is taken care of by the proper sizing of the gear blank.

While it may be customary practice to make the hob an exact duplicate of the worm except in the one item of outside diameter, a hob proportioned as suggested in Fig. 7 is recommended as one that will give much more satisfactory results, and be found to be well worth any additional trouble in construction required beyond that for the style ordinarily used. The peculiar feature of this hob is that it is an

* MACHINERY, September, 1907.

exact opposite of the worm with respect to the proportions of the thread shape; the depth below the pitch line in one case being equal to the height above the pitch line in the other. The object of this is to have a hob that will form the complete outline of the tooth and make it absolutely certain that the standard proportions of tooth and clearance are obtained. Thus, should the diameter of the blank be large, the hob will trim off the top of the gear teeth to the proper length, when the proper center distance is maintained.

There is another point that is generally overlooked, and that is the necessity for having the corners of the thread rounded over, and for providing a liberal fillet at the root of the thread. The radii of the rounded corner and the fillet may be as large as the clearance will allow, which would be one-twentieth of the circular pitch of the thread.

The effect that this fillet and rounded thread have on the shape of

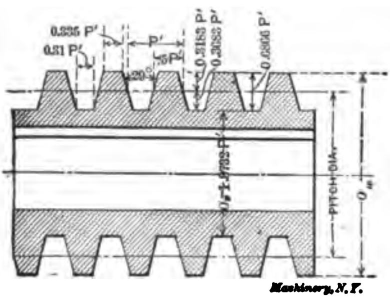

Fig. 6. Dimensions of Worm

the tooth is something that greatly increases the quality of the gear and the strength of each individual tooth. The rounded corner on the thread points does away with any tendency to scratch the surface of the tooth in the cutting action, and leaves a much larger fillet at the root, greatly increasing the strength. The fillet at the bottom of the thread rounds off the top of the tooth in the worm-gear, removing any burrs, and leaving a nicely finished product. This fillet also removes the dangerous tendency of the hob to develop cracks in the hardening process—a common source of trouble even where care is taken. Fig. 6 shows the proportions of the worm in comparison with the hob in Fig. 7.

In forming the hob, much can be gained by making a special form tool of correct proportion that will leave no chance for error; the only dimension needing care then, is the diameter. Such a tool is shown in Fig. 9. The figure is dimensioned by formulas, so that a tool for any pitch can be easily proportioned from it. This tool may be made by using a gear caliper without resorting to the protractor, or the protractor may be used in laying out the angle. This tool may be made without side clearance, providing that the sides incline in the same direction and at the same angle that the thread takes, but under ordinary circumstances, where only one hob is to be made, little is gained by having no side clearance. Clearance may be made

from 5 to 10 degrees from the angle of the thread. Grinding a tool like this of course changes its form, so it must not be used indefinitely in making large numbers of similar hobs.

Number of Flutes in Hobs

The number of flutes that should be provided in the hob is a point on which very little is said, various authorities differing widely. Where the hob is to be used in an automatic hobbing machine in which the hob and blank are positively geared together, the number of flutes may be a comparatively small number as compared with a hob that is to be used in connection with ordinary processes of hobbing worm gears. In the process in which the previously gashed worm-gear blank is swung loosely on centers and revolved by the hob as the latter rotates, the hob should have a larger number of flutes.

A rule that checks up well with present practice is as follows:

To find the number of flutes in a hob, multiply the diameter of the hob by three, and divide by twice the circular pitch.

The above rule gives suitable results on hobs for general purposes.

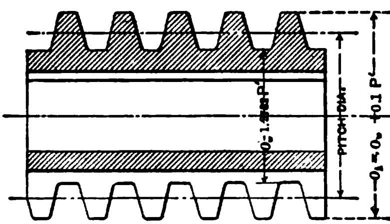

Fig. 7. Dimensions of Hob

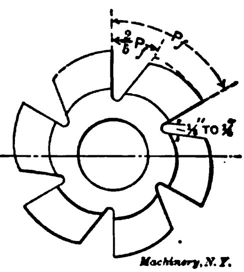

Fig. 8. Data for Fluting Hob

When the result gives an odd number of teeth, take the next smaller even number, to facilitate calipering.

Some authorities on worm-gearing state that the number of flutes in a hob should in no case be an exact multiple of the number of threads. Their reason for this rule is that the hob so gashed will produce a much smoother tooth and one nearer correct in shape, because no tooth in the hob passes the same tooth in the gear twice in succession, so that any little imperfections in shape of the individual hob teeth are counteracted by one another. Another authority is strong in his advice not to have the circumferential distance from flute to flute equal to or equally divisible by the circular pitch, for the same reason as stated regarding the former rule. From these statements, it is seen that to obtain a rule that would be at once simple and yet take all conditions into consideration, would be a difficult proposition. It seems, however, that only the first of these two rules is a logical one. Owing to the fact that hobs have teeth only, instead of full surfaces matching the worm, the curved outlines of the wheel teeth are merely approximated by a series of tangents. If the number of flutes in the hob is a multiple of the number of threads, the hob teeth will "track" after each other, giving wheel teeth

only roughly approximated by a comparatively small number of long tangents.

The cutter used in gashing the hob should be about ⅛ inch thick at the periphery for hobs of ordinary pitch, while for those of coarser pitch a cutter ¼ inch thick would be much better. The width of the gash at the periphery of the hob should be about two-fifths the pitch of the flutes. The cutter should be sunk into the blank so that it reaches from 3/16 to ¼ inch below the root of the thread. Fig. 8 shows an end view of a hob gashed according to these rules.

Where a hob is to be used to any great extent, and is subject to much wear, it would be advisable to increase the diameter above the

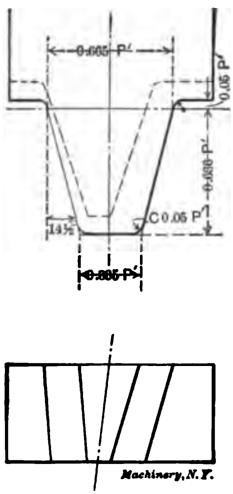

Fig. 9. Dimensions of Tool for Threading Hob

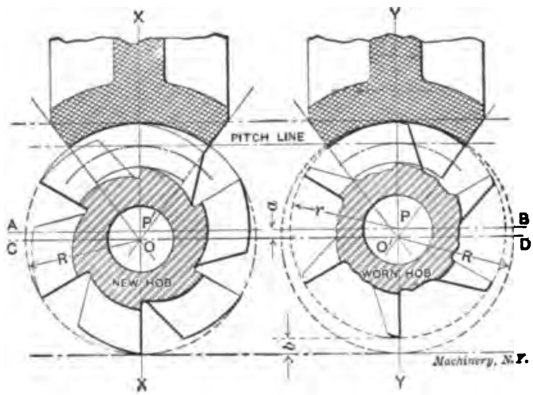

Fig. 10. The Difference in Shape of Teeth Cut by New and Old Hobs

dimensions given from 0.010 to 0.030 inch according to its diameter and pitch, to allow for decrease in diameter due to the relief, and caused by grinding back the cutting face in sharpening.

Hobs are generally fluted parallel with the axis, but it is obvious that they should be gashed on a spiral at right angles with the thread helix in order that the cutting face may be presented with theoretical correctness; but the trouble encountered in relieving the teeth on the ordinary backing off attachment is the cause of the common mode of fluting. When the pitch or lead is coarse in comparison with the pitch diameter of the hob, so that the angle is correspondingly steep, it may be best to flute on the normal helix, and if the hob cannot be machine relieved, it may be backed off by hand.

The amount of relief depends much on the use for which the hob is intended. A hand hob for hobbing a gear in position may be made with little or no relief, while hobs used on hobbing machines may have much more relief than those used on the milling machine.

CHAPTER III

SUGGESTED REFINEMENT IN THE HOBBING OF WORM-WHEELS*

At the left of Fig. 10 is a sectional view showing a hob in the act of putting the last finishing touches on a worm-wheel. The hob is supposed to be a new one and is shown in the condition it is in when first received from the makers. At the right of Fig. 10 is shown the same hob putting the finishing touches on a worm-wheel similar to that in the first case. The hob in this case is represented as having been in use for a considerable time, and having been ground down to the last extremity, ready to be discarded for a new one. A study of this cut will show that if the hob is made in the first place to properly match the worm which is to drive the wheel, it will not, when worn, cut exactly the proper form of tooth in the blank to mesh with that worm. The teeth are cut to the same depth in each case, this being necessary in order to make a proper fit with the worm, which is the same in each case and is set at the same center distance. The grinding away of the worn hob has reduced its diameter by an amount indicated by dimension *b*. Its center is therefore at *P* on the line *A B*, which is offset by a distance represented by dimension *a* from the line *C D* on which the center *O* of the new hob is located. This reduction in diameter as the hob is ground away from time to time, so evidently follows from the construction of the relieved hob, that it scarcely needs to be explained.

It is said of relieved hobs that they can be ground without changing their shape. This is true so far as the outline of the cutting edge is concerned, but it will be evident on examining the conditions shown at the right hand of Fig. 10, that whatever the outline of the cutting edges, a new hob of radius *R* will not cut exactly the same shape teeth in the blank as the worn hob with radius *r*. The elements of the tooth surface it generates are struck from a center *P*, removed by dimension *a* from center *O'* which is the location of the axis of the worm with which it meshes.

It is possible, and perhaps practicable, to overcome this slight error; that is, to so design and use the hob that it will cut as correct teeth when worn as when new. In Fig. 11, dotted line *A A* represents the outlines of a new hob in the act of finishing the worm-wheel shown. Were a hob, ground as shown at the right of Fig. 10, to be substituted on the arbor for this new hob, without altering the adjustment of the machine except to move the hob endwise and bring it in contact with the teeth of the wheel on one side, this hob would be represented in Fig. 11 by the full line *B B*. It is evident that the left-hand cutting edges of this hob coincide (to the depth they extend into the wheel) with those of the new hob represented by outline *A A*. They will,

* MACHINERY, May, 1907.

therefore, so far as they extend, cut identically similar and correct tooth curves with the new hob.

Teeth cut with this worn hob would, however, evidently have two faults. The space would be too narrow at the pitch line by a distance measured by dimension *m*, and they would not be cut deep enough in the blank by a distance measured by dimension *n*. Our problem is to so alter the design and application of the hob, that, even when worn, we can cut the teeth deep enough and the space wide enough.

Fig. 12 shows these conditions fulfilled. Dotted line *C C* shows the outline of the proposed hob when new. The only difference between the proposed hob and the regular one, whose outlines are shown by the dotted line *A A* in Fig. 11, is that the teeth have been lengthened by an amount equal to dimension *o*. The hob is fed in as was the case with the new hob in Fig. 11 until the distance between its center line and that of the blank is the same as that between the center line of the worm and the wheel in the finished machine. The increase in

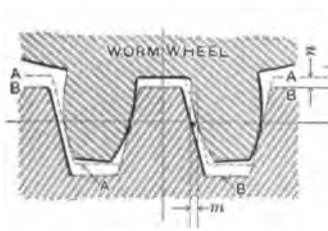

Fig. 11. Cutting Action of Ordinary Hob at Fixed Center Distance, when New and when Worn

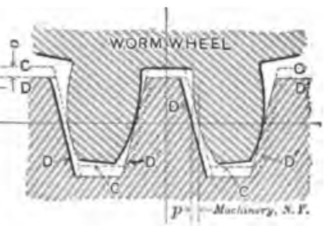

Fig. 12. Cutting Action of Proposed Hob, when New and when Old

radius, then, by an amount *o*, makes the hob cut a clearance deeper than is necessary by that amount. In a spur gear this would doubtless be a bad thing, since it would make the tooth slenderer and therefore weaker. A worm-gear, however, if designed to be sufficiently durable for continuous use, is almost certain to be several times stronger than necessary, so that the slight weakening involved in the change is not of great importance. When the hob is worn to the shape shown by the full outline *D D*, the hob is evidently of the same diameter as the new one in Fig. 11, represented by dotted outline *A A*. Our tooth space, however, as before explained, will be too narrow by the amount *m* in Fig. 11 or *p* in Fig. 12. To widen it out sufficiently, it is therefore necessary for us, after the hob has been fed in to the proper depth, to still continue the cutting action, feeding the hob endwise, however, until it has been displaced to the position indicated by outlines *D′ D′*. The resulting tooth is evidently identical with that given by the new hob *A A* in Fig. 11.

It will be understood that when the hob in Fig. 12 is new, it will not have to be shifted end-wise at all, since it will cut a tooth space of the proper width as soon as fed to depth. It will, however, cut a space deeper than necessary by an amount *o*. The worn hob, on the other hand, has to be shifted longitudinally by an amount *p* and cuts to exactly the required depth. These represent the two extreme con-

ditions. When the hob is half worn, the excess clearance will be equal to half of o, and the longitudinal displacement necessary will be equal to half of p.

While the change in the design of the hob could be made easily enough, there is doubtless some difficulty in making the required change in the hobbing of the blank. Taking it for granted that the hob has been made to suit the worm which is to be used, and that it, therefore, has the same pitch diameter and thickness of tooth at the pitch line, the method of procedure will invariably require that the hob be fed in to the worm-wheel blank until the distance from the center of the hob to that of the wheel is the same as the distance from the center of the worm to that of the wheel in the finished machine. This will be true whether the hob is new or worn, and whatever may be the kind of machine on which the hobbing is done.

The method by which the hob is displaced longitudinally will depend on the machine used for the operation. There will be no possible way of doing it if the wheel is being finished while running loosely on centers, as is common practice when the blank has first been gashed. It is required that the hob and blank be positively geared together. If a positively driven hobbing attachment in the milling machine is being used, the matter is simple. If the hob is being driven by the spindle of the machine, throw in the cross feed in either direction until the required longitudinal displacement of the wheel with relation to the hob has taken place. The question as to when this has taken place may be decided either by measuring the thickness of the tooth, as in cutting spur gears, or by trying the wheel from time to time with its worm, the two parts being mounted in place in the machine they are to go in, or held the proper distance apart by other means.

For regular hobbing machines, as at present made, the matter is more difficult. The required longitudinal displacement of the hob may be obtained, in effect, by a rotary displacement of the hob which may be accomplished by slipping (a tooth at a time), the teeth of the change gears connecting the hob and the blank. If a hobbing machine were to be built especially for use in the way which is here suggested, differential gearing could be introduced in the train between the hob and the wheel, to which a power feed could be given to effect the rotary displacement when the hob has been fed to depth, or a power feed might be applied to feed the spindle and its attached hob endwise to effect the same result.

It is not certain that the error which exists in the use of relieved hobs is of enough importance to warrant taking any trouble to remedy it. It is always well, however, to know and understand such errors as may exist in any process of this sort, no matter if they are of no great practical importance. While some designers and shop men have doubtless recognized the existence of this particular error, still probably most of them take it for granted that the process is absolutely accurate, since they are so often reminded that the relieved hob can be "ground without change of shape."

CHAPTER IV

THE LOCATION OF THE PITCH CIRCLE IN WORM GEARING

Different authorities and writers on mechanical subjects have always held very different opinions regarding the location of the pitch circle of a worm gear. No better example of these differences in opinion can be given than by repeating a discussion in relation to this interesting subject which took place in the columns of MACHINERY, during 1905. The subject was brought up by Mr. Oscar E. Perrigo, who, in describing the feed arrangement of a heavy turret lathe, into the design of which the worm and worm-gear entered, found occasion to state his opinions in regard to the construction of this mechanism. Mr. Perrigo says*:

"Many good mechanics are so prone to object to any kind of a worm-gear, and can cite numerous examples wherein they have proven failures and utterly worthless for the purposes intended, that there is a very strong prejudice against them in any form. The writer is of the opinion that there is really only one practical objection to a properly constructed worm-gear, and that is, it must be constantly lubricated, and men running machines in which they are used are very liable to forget this fact altogether. The principal, and almost the only reason why worm-gears fail to give satisfactory results is that usually they are not properly designed at first. Another is that they are not properly hobbed out, and sometimes not hobbed at all. It is the purpose of this article to point out how they should be designed in order that they may be successful.

"There are various methods for determining the diameter of the pitch circle of a worm-gear. One authority takes the outside diameter of the turned blank at its smallest diameter, or throat, as proper. Another takes the diameter of the bottom of the teeth at the extreme edge of the cut gear; still another, the point where the pitch line of the worm intersects the center line passing through the worm and worm-gear. All these are more or less in error, as they do not take proper account of the width of the face of the gear. If the teeth are straight, as in a spur gear, we naturally take a point in the center of the teeth (after subtracting the clearance) as the pitch line. Now when we have a curved tooth it obviously is not proper to do this, as the actual working pitch diameter must be somewhat larger than this; but how much larger should evidently be determined by the amount of contact with the worm, that is, the angle within which this contact is to be, the width of face being in turn controlled by the diameter of the worm.

"Practically, the face of the worm gear is about equal to one-half

* MACHINERY, June, 1905.

the outside diameter of the worm, but the matter is best considered by saying that the enclosed angle of contact should not be less than 45 degrees nor more than 80 degrees, while from 60 degrees to 70 degrees will be found most useful. The writer has found by ample practice that the true working pitch diameter is most nearly determined by the method shown in Fig. 13, which represents a worm-wheel having a contact of 70 degrees. To determine the pitch diameter, divide the arc of the pitch line of the worm, contained between the center line and one of the lines of the enclosing angle, into three equal parts, and draw the line *a* at the intersection of the second line from the center line. This will give the point from which to measure the pitch diameter. If this is laid out on a large scale and with various angles of contact, the difference between it and the usual methods will be more clearly shown than it is in the engraving." It will be found to make a difference of several teeth in a worm-wheel of a fairly large number of teeth.

As to the proof of the correctness of this method of designing worm-gears, Mr. Perrigo states that he has used it successfully for years.

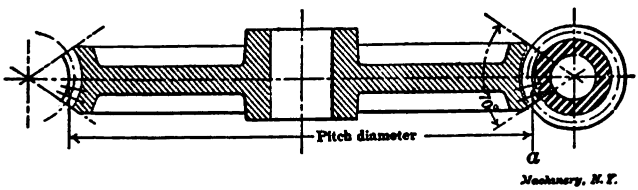

Fig. 13. Method of Determining the Pitch Diameter of a Worm-gear

The turret lathe, previously referred to, on which this worm gearing acted as a drive for the feed, would readily bore 3-inch holes in 50-point carbon steel spindles. In several cases where a 5⅛-inch hole was required, it was first bored 2 inches and then a boring bar, provided with two double-end cutters, was introduced, enlarging the hole from 2 inches to 5⅛ inches at one cut and taking out nearly thirty pounds of chips per hour. The machine had been in use for over seven years, and the same worms and worm-gears were on it that were put on when the machine was first built, and they were in good condition for as many years more of good service. The working faces did not seem to have changed their original form during the entire time, which, Mr. Perrigo says, may be taken as ample evidence that they were right originally, particularly as he had frequently seen worm-gears in lathe aprons, designed after the usual methods, entirely worn out with six or eight months' service.

Undoubtedly prompted by Mr. Perrigo's statements in regard to the worm-gear, Mr. John Edgar, a few months later,* added to the discussion on the subject. He mentions first the method for the location of the pitch circle accepted as standard practice. According to this method the pitch line of a worm is located on a circle whose radius is smaller than that of the worm by an amount equal to one-half the

* MACHINERY, October, 1906.

working depth of the tooth. Where the working depth, as in standard practice, is equal to 0.6366 times the linear pitch, and when P' is the linear pitch, o the outside diameter, and d the pitch diameter of the worm, this fact may be expressed by the following formula:

$$d = o - 0.6366\,P' \qquad (1)$$

In Fig. 14 we have a section through a worm and worm-gear. The pitch circle for the worm, according to standard practice, is located as shown tangent to the line E, which is the pitch line of the worm-

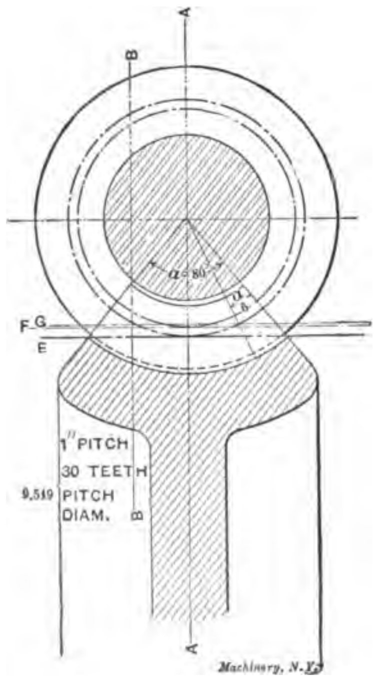

Fig. 14

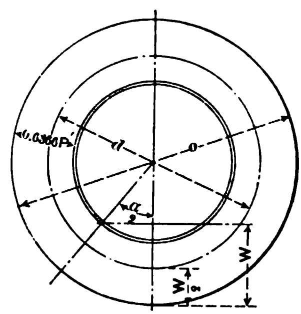

Fig. 15

gear. On inspection of the figure it is seen that while the addendum of the worm and worm-gear are equal at the center line AA, they are not at any other point along the pitch line, either to the right or the left. A section taken through the gear on the line AA would reveal teeth similar in shape to those of a spur gear of the same pitch and number of teeth. But how does this shape of the teeth vary as we shift this line either side of the central position? Let us show this by an example, taking the case of a worm having a single thread of 1-inch pitch. By taking a section on line BB instead of the center line AA we obtain Fig. 16. This figure shows plainly that the faces of the teeth of the gear are considerably longer than the flanks. It is easily seen that the greater the angle a is, the greater will this difference be, and *vice versa*, until we reach the central position, where there is

no difference. Therefore we see that this angle a plays an important part in the design of a successful worm-gear.

This angle is not the only cause of distortion in the shape of the tooth. With a little thought it will be seen that the angle of the helix also is a cause for further irregularity. To illustrate this we will take

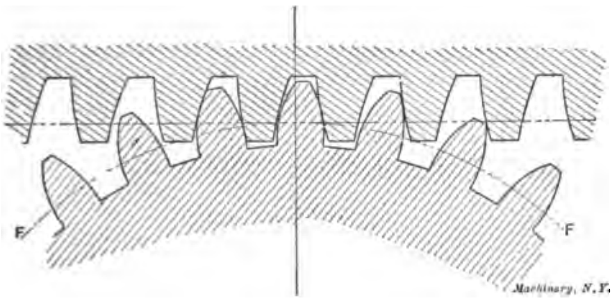

Fig. 16. Section at Line BB, Fig. 14, Single-thread Worm, One-inch Lead

the case of a worm having the same pitch, but having three threads instead of one, giving a lead of 3 inches. A section of this at BB is shown in Fig. 17. These conditions have the effect of producing even longer faces than do those in the former case.

What can be done to remedy this defect? We can shorten the faces, but when we do that at this point we do so all along the face of the gear and thus change the shape at AA, where it is normal. Therefore, the best we can do is to divide the difference at the two extreme points—AA and BB. This can be done as follows: In an ordinary spur gear of standard proportions the pitch line is located at a point midway of the working depth. From Fig. 15, which shows the end

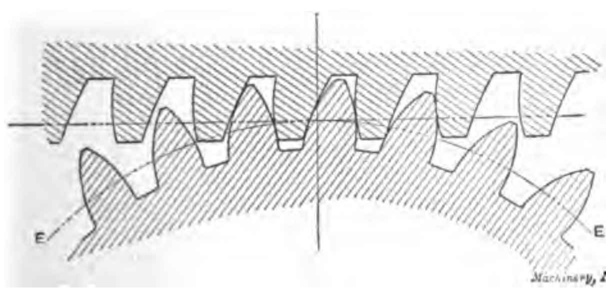

Fig. 17. Section at Line BB, Fig. 14, Triple-thread Worm, Three-inch Lead

view of a worm, we see that the total working depth is equal to W, so that from the foregoing statement the pitch line should pass through a pointed situated at a distance equal to one-half of W from the outside of the worm, making d the pitch diameter of the worm.

By an inspection of Fig. 15 we may derive the following formula:

$$W = \frac{o}{2} - \cos\frac{a}{2}\left(\frac{o}{2} - 0.6366P'\right) \qquad (2)$$

Since $d = o - W$, we may obtain the value of d in terms of o, P' and a:

$$d = \frac{o}{2} + \cos\frac{a}{2}\left(\frac{o}{2} - 0.6366\,P'\right) \qquad (3)$$

Solving this last equation for o, we have the means for finding the outside diameter when d, P' and a are given:

$$o = \frac{2\,d + 1.273\,P'\cos\dfrac{a}{2}}{1 + \cos\dfrac{a}{2}} \qquad (4)$$

Formulas (3) and (4) may be for obtaining the pitch diameter of any worm when the outside diameter is known, and *vice versa*.

It is quite evident, says Mr. Edgar, that the method given by Mr.

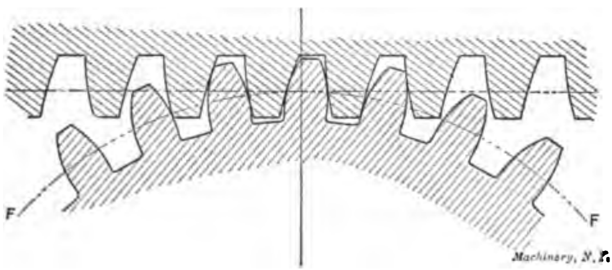

Fig. 18. Section at Line BB, Fig. 14. Pitch Line Determined by Formula (3)

Perrigo for obtaining the pitch diameter of the gear is based on this principle, but it is only an approximation, the variance between its results and those of the formula increasing with the angle a. The difference for the example we have been investigating will be seen in Fig. 14 where G is the line as located by his method, F that by the formula, and E the standard location.

To show the difference this change in location of the pitch line makes in the tooth shape as compared with the usual practice, sections have been drawn at *BB* for both a single- and a triple-threaded worm of 1-inch pitch. Figs. 18 and 19, respectively, show these sections. Here we see that while the faces are yet considerably longer than the flanks, the shape is improved. The difference between Fig. 18 and a normal section is very slight and hardly noticeable, and while the shape in Fig. 19 is somewhat freakish, it has all the properties of a smoothly running gear.

But someone may ask what all this has to do with the durability of the gear. It is this: It has been proved that the friction of approach is much more in amount than that of the release. This friction of approach occurs between the face of the driven gear and the flank of the driver. Now if these particular elements of the tooth are extra long, the friction is proportionately increased over what it

would be in a normal tooth. The friction of motion is always accompanied by wearing of the surfaces in contact; therefore in order to increase the life of the gear, we must decrease the friction to a minimum. This we have done by locating the pitch line in accordance with the formula.

In order to illustrate the extent to which some designers go to eliminate the friction between the surfaces of the teeth in contact, the case of some special forms of clock gearing may be cited where the driver is made with teeth having no flanks and the driven gear with teeth having no faces, fixing all the contact at the period of release. The importance of this point is easily ascertained by observing the wear on the teeth of a pair of gears that run constantly in one direction.

The tooth curves in the above figures were obtained by the tracing cloth method described in Unwin's "Machine Design." The subject in

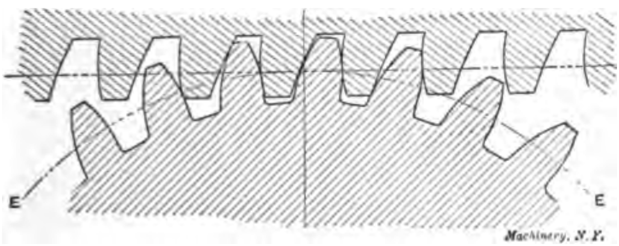

Fig. 19. Section at Line BB, Fig. 14. Pitch Line Determined by Formula (8)

hand, however, does not require or warrant the description of this method here.

Finally, Mr. Ralph E. Flanders added to the discussion by a more fundamental study into the principles involved than had been undertaken by any of the previous writers. His analyzation of the subject clears some of the doubtful points at issue. In order to give a comprehensive idea of his statements, his treatment of the question has been given verbatim in the following*:

On the Location of the Pitch Circle in Worm Gearing

Mr. Perrigo and Mr. Edgar, in their recent contributions on this subject, have called attention to some important points in connection with this form of gearing. The writer feels, however, that the recommendations they make cannot be followed blindly, but must be applied with a full knowledge of the limitations within which these recommendations are useful. It is the purpose of the present article to point out these limitations.

Mr. Perrigo describes a worm and a worm-wheel which he has incorporated in the feed mechanism of a screw machine. Made in the way he describes, this worm and wheel have outlasted everything of their kind in his previous experience, and if the cases with which he mentally compares this one have no other important points of difference,

* MACHINERY, November, 1905.

his confidence is certainly justified. Unfortunately, this point is not covered, and so we are left without a solid foundation on which to base our judgment.

The feed worm of a screw machine, if it is of the class in which the worm is dropped out of engagement when the feed is released, does its work under peculiarly trying circumstances. The writer's experience in screw machine design has led him to believe that the proper proportioning of these parts is a matter of considerable importance. Consider the case of a bronze wheel and a hardened steel worm working under the pressure of a heavy cut: When the worm is released from engagement with the wheel, under the pressure of this heavy cut, the sharp, hardened corner of the worm-tooth goes sliding down the face of its corresponding tooth in the wheel, giving it a last dig as it jumps by the corner. The necessity for quick handling

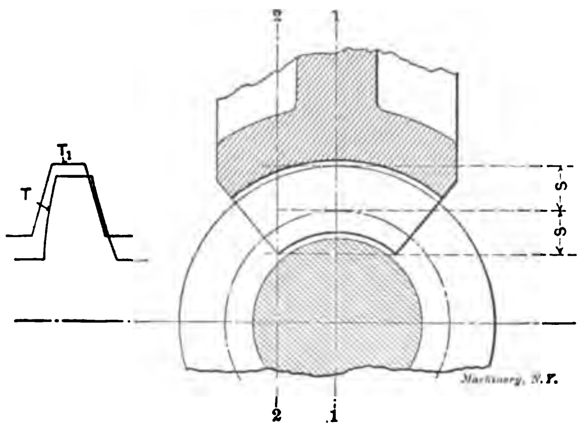

Fig. 20

demands that the momentum of the revolving parts of the feed mechanism be kept as low as possible, so the peripheral speed of the worm-wheel must be as low as possible in comparison with the rate of movement of the slide. This, in turn, requires the worm to work under heavy pressure. It is not practicable to locate the feed release between the worm and the clutch, especially if the feed is to be stopped automatically, because it is difficult to handle a toothed clutch under a severe torsional strain. Usually this problem is settled by a compromise, the success of which depends on the designer's judgment; the peripheral speed of the worm-wheel is made as high, and consequently, the worm thrust is made as low as is possible without too great a sacrifice in rapidity of handling. In large machines this difficulty may be overcome by connecting the pinion shaft to the worm-wheel by frictional contact, accomplished by tightening up a supplementary pilot mounted in front of the main pilot wheel; the automatic release is effected by stopping the rotation of the worm.

Another point that militates against the durability of this mechan-

ism when a releasing worm is used, is the indeterminate location of the worm. While it is obvious that a worm cannot be adjusted in a direction parallel to the axis of the worm-wheel, it is not generally realized that the center distance between its axis and that of the wheel cannot be varied without losing the perfect action which exists when the worm is properly located. That this is so will be evident from Fig. 20. In this cut T_1 and T are sections of a worm tooth taken on lines 1-1 and 2-2 respectively. The section on 1-1 is evidently that of an involute rack tooth and so possesses the characteristic property of correct action at any center distance, so long as its straight face is in contact with the mating gear tooth. As we leave this section, however, and approach section 2-2, the tooth outline gradually loses its resemblance to the involute form and takes a shape in which positive location is absolutely necessary for correct action, as is shown by the curved sides. This variation from the

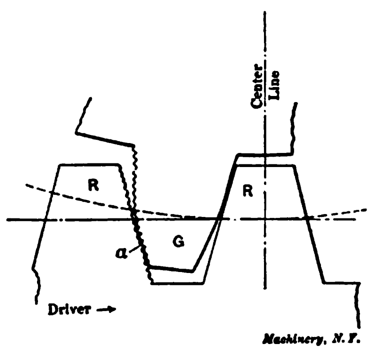

Fig. 21

involute shape is especially marked in worms of large helix angle and consequent high efficiency.

Now, if the worm is slightly separated from its correct location in the mating wheel and no sideways motion is allowed, it will be seen by observing the relative angularity of the outlines of the faces in the curves T and T_1 that the contact will at once lose its character of line contact, extending across the full width of the gear, and will be concentrated in point contact on the extreme outer edge, where correct action is impossible except at the calculated center distance. For working under heavy pressure, then, it is necessary that the worm agree in shape with the hob which cut its mate, and that its axis exactly coincide with that of the hob when this was taking its finishing cut. These requirements may be met easily in high-grade work, such as is the rule in making a worm-gear drive for a gear-cutter spindle or an elevator, but such workmanship is very far from the haphazard fitting that a releasing feed worm must necessarily get.

It has occurred to the writer that the worm, or worms, in Mr. Perrigo's turret lathe, must be of considerably greater helix angle than is usual in feed gearing. The unusual arrangement of a double reduc-

tion is employed, making use of two sets of worms and wheels in series. Unless the feed shaft rotates at high speed, or the feed is exceedingly fine, this must mean that the reduction in each set of gears is small, which in turn predicates a large helix angle and an efficient gear. Mr. Perrigo must, then, give us more definite information if his experience is to be valuable as a permanent record in the matter of the location of the pitch line. Was his machine furnished with a releasing worm for a feed stop, and were the machines with which he compares it so equipped? How carefully was the worm

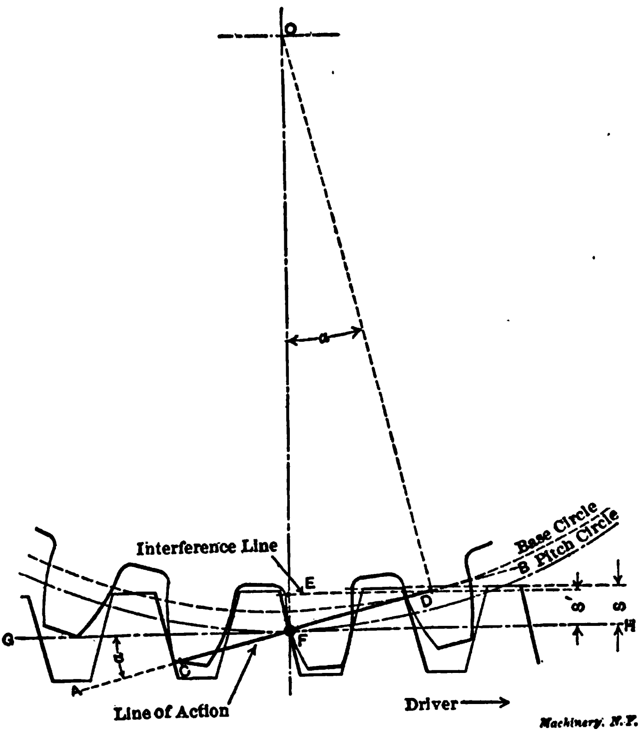

Fig. 22

fitted in the last machine and in the former machines? What are the helix angles of these worms and former unsuccessful cases? What materials were used in the different sets of gears which are under comparison?

Mr. Edgar has shown quite plainly that the advantage to be gained by lessening the diameter of the pitch circle on the worm is due to the fact that in such a case the contact between worm and wheel takes place for the most part after the teeth have begun to recede from each other. In Fig. 22 the worm, with its pitch line at *GH*, driving the wheel in the direction shown, will always make contact with it along the line of action, *CD*. The pitch line is located, as usual, half-

way down the working depth of the tooth, and as may be seen, the contact is almost equally divided on each side of the center line. In Fig. 23, with the same reference letters, the pitch line has been located according to the rule proposed by Mr. Edgar, and the contact between the teeth is seen to take place almost wholly during the time when they are leaving each other.

Friction between two rubbing surfaces is due to the resistance imposed by the microscopic irregularities which exist on even the smoothest surfaces. In Fig. 21 are shown two teeth approaching each other, in which these irregularies are greatly exaggerated. *R* is the driving and *G* is the driven tooth. Evidently if these irregularities were as great as shown, the teeth would lock together and movement would be impossible; on the other hand, if *G* were the driving tooth, and the teeth were separating, there would be little to hinder their free movement. It is, then, desirable that most of the contact should take place when the teeth are leaving each other, to avoid friction, loss of power, and wear of tooth surfaces.

Fig. 20 shows the way in which Mr. Edgar proposes to locate the pitch circle of the worm. This circle is tangent to a line which lies at equal vertical distances from the extreme working points of the worm-wheel tooth, and he locates the pitch line here because it is so located in a spur gear. To the writer it seems that there is no analogy between them. The pitch line of a spur gear is located at one-half the working depth of the tooth because it is required that a set of standard spur gears be interchangeable, a gear of any number of teeth meshing perfectly with a gear of any other number of the same pitch. This requirement is entirely outside of the sphere of worm gearing, so we may locate the pitch line at any point that will give favorable results as regards efficiency and durability.

The location of the pitch line affects the working qualities of the gearing in four ways, at least. With a worm of given diameter and pitch, and a wheel of given number of teeth and angle of contact, it determines the effective working area of the teeth in both members, the strength of the teeth in the wheel, the number of teeth in contact, and the nature of the contact, that is, whether it takes place during the approach or the release.

Fig. 22 shows a central section of a worm and wheel calculated in the usual manner. If a is equal to the pressure angle, and angle *FDO* is a right angle, a circle drawn from center *O* through *D* will be the base circle from which the involute curve is formed, and the line of action—the line in which the working contact between the teeth will take place—will lie in line *AB*. This line of action will evidently be limited at one end by *C*, the point where it crosses the outside diameter of the wheel at its throat, and at the other by *D*, the point where line *AB* is tangent to the base circle, since the involute does not extend inside of the circle from which it is derived. It is plain, then, that all that part of the wheel tooth which lies inside of the base circle is clearance, and unfit for bearing surface, and that all of the worm tooth which extends above point *D*, or the "interference line," as it is

marked, serves no useful purpose. This area of the worm tooth extending above the interference line, is seen to be slight for a thirty-tooth wheel of standard design. An inspection of the cut will show that there are always two and sometimes three teeth in contact. The contact takes place about equally each side of the center line, inclining toward the favorable side, since line *FD*, on the release, is somewhat longer than *CF*, on the approach.

In Fig. 23, we may see what effect has been produced by increasing the addendum of the worm, as we are advised to do. In the first place,

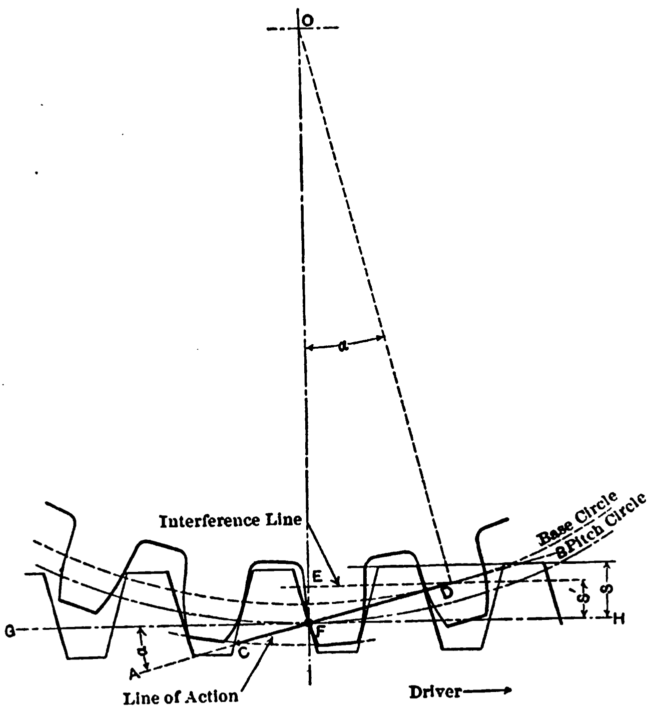

Fig. 23

the interference line, through point *D*, is lowered so far that the useful area of the teeth has been greatly decreased. There is no working contact on the wheel teeth inside of the base circle, nor do the worm teeth serve any useful purpose above the interference line.

The strength of the wheel tooth has been decreased. The great length of useless worm tooth extending above the interference line has cut a deeper clearance into the flank of the wheel tooth, thus weakening it at the very point where it needs strength. On pages 68, 69, and 71, of the Brown & Sharpe "Treatise on Gears," are three illustrations which show this point very clearly. One case is that of a twelve-tooth worm-wheel of standard design, which gives a badly

undercut flank. In the next illustration the worm-wheel has been hobbed according to Mr. Edgar's rule, and the result is worse than in the first case. The last illustration shows the pitch line thrown clear to the outside diameter of the worm, this being advised as the proper remedy to secure a tooth of sufficient strength and bearing surface.

Referring again to Fig. 23, the number of teeth in contact has been reduced until there is only one constantly in use, though two are in position to work most of the time. The single gain to be derived in return for the advantages that have been lost, lies in the fact that a greater percentage of the line of action lies on the releasing side of pitch point F than before, since FD is noticeably longer than FC.

Of course only the action on the center line has been analyzed. The writer has studied the action at sections made in different places in the worm-wheel face, and it looks as if the conditions at the center line were a fairly good index of what is going on nearer the sides. The line of contact appears to rise slightly toward the outside of the worm as it leaves the center (going toward the leading side of the worm), and then drops again toward the edge of the wheel. On the retreating side of the worm the contact drops continuously. This tends to minimize the effect that the width of the wheel has on the action.

How, then, should the pitch line be located? It seems to the writer that the problem is so involved that in a case of any importance the designer should not trust to any empirical rule, but should plan each case with reference to these four points: area of bearing surface in the teeth, strength of the teeth, number of teeth in contact, and location of contact, whether in the approach or the release. To these should be added a fifth point, more important than any of the others, as far as efficiency is concerned, and that is in relation to the helix angle of the worm: it should be as large as possible.

Taking all these points into consideration, it would seem that, for worms and wheels made as they usually are for ordinary service, from ordinary materials, and with ordinary carefulness of workmanship in making and fitting, it is hardly worth while to bother about changing the location of the pitch line for the sake of having the contact on the release. It introduces too many other complications into the problem. Still, if there is any one who wants to try the effect of altering the worm and wheel dimensions with this end in view, here are a few suggestions in the shape of formulas to add to those of the two contributors who have previously written on this subject.

Let N = number of teeth in wheel.
O = throat diameter of wheel.
P' = linear pitch of worm.
o = outside diameter of worm.
D = pitch diameter of wheel.
d = pitch diameter of worm.
a = pressure angle.

$$C = \frac{D + d}{2} = \text{center distance between the worm and the wheel.}$$

S' = effective height of worm tooth above pitch line (see Fig. 23). An inspection of Fig. 23 will show that S' may be expressed as follows:

$$S' = \frac{D \sin^2 a}{2}$$

If we limit the height of our tooth to this line, thus allowing no interference, we may use the following formulas, it being considered that we have given C, P' and N.

$$D = \frac{N P'}{\pi} \tag{5}$$

$$d = 2C - D \tag{6}$$

$$o = d + D \sin^2 a \tag{7}$$

$$O = D + 1.273\,P' - D \sin^2 a \tag{8}$$

For a pressure angle of 14½ degrees and an allowed interference equal to that of a standard worm in mesh with a 25-tooth wheel, these last two formulas will become:

$$o = d + \frac{D}{13} \tag{9}$$

$$O = 0.923\,D + 1.273\,P' \tag{10}$$

These formulas will give as much of the contact on the release as is possible without too much undercutting; the location of the pitch line will, of course, vary widely. Formulas (7) and (8) (when a = 14½ degrees) are good for any number up to 64 teeth, and Formulas (9) and (10) up to 52 teeth. Above these numbers the formulas would bring the pitch line below the root diameter of the worm, which is needless; so for such cases, Formulas (7), (8), (9), and (10) should be replaced by the following, which will keep the pitch line within the working area of the tooth:

$$o = d + 1{,}273\,P' \tag{11}$$

$$O = D \tag{12}$$

All that has been said in the preceding paragraphs refers only to worms whose tooth outlines show straight sides on an axial section. If, as is often the case with steep-pitched worms, the cutting tool is made with straight sides, but tipped up at an angle to agree with the helical angle of the worm, an axial section will show teeth with curved sides whose shape will depend upon the helical angle. In such a case as this it is impossible to apply any of the rules which govern the action of involute teeth, and the only way to go about the matter of locating the pitch line to suit the ideas of the designer is to make a careful analysis of the tooth action on various sections. This operation would be so troublesome and tiresome as to be impracticable under any ordinary circumstances.

CHAPTER V

THE HINDLEY WORM AND GEAR*

The Hindley type of worm-gear was first used in Hindley's dividing engine,† and was, by the inventor, considered superior to the ordinary type, in wearing quality. Investigation has practically settled that the nature of contact between the worm thread and the teeth of the ordinary worm-wheel is that of line contact, extending across the tooth on the pitch line. It has also been fairly well proved in practical examples that the contact is of a broader nature on account of the elasticity of the materials used in the construction. The convex surfaces of contact are flattened considerably under pressure and thus for practical purposes make actual surface contact. The contact in the ordinary worm and worm-wheel type is limited to two teeth of the wheel and worm thread, at most.

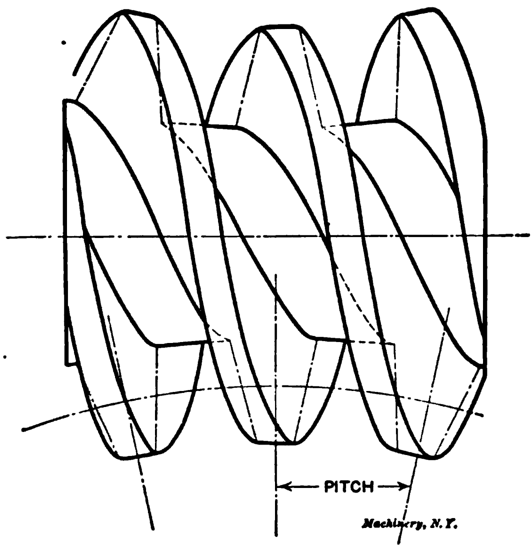

Fig. 24. Typical Hindley Worm

Comparison of Ordinary and Hindley Worm Gearing

The conditions are much different in the case of the Hindley worm, and it is the intention in this chapter to show wherein the difference lies. As this style of gearing is uncommon to most of us, a few words

* MACHINERY, December, 1908.

† The Hindley gear, as used in the Hindley dividing engine, is described by Smeaton, also by Willis (Principles of Mechanism, 1851). Various modifications of the Hindley gear, including Jensen's winch, are illustrated in Reuleaux's "Constructor," page 143.

regarding its construction will not be out of place. Fig. 24 illustrates the Hindley worm, showing the theoretical form. This worm is not of cylindrical shape, but is formed somewhat like an hour-glass, after which it is sometimes named. The worm blank, being made smaller in diameter in the middle than at either end, conforms to the circumference of the wheel with which it meshes. The worm thread is cut by a tool which moves in a circular path about a center identical with the axis of the wheel with which it is to mesh, and in the plane in which the axis of the worm lies. The process is similar to ordinary thread cutting in the engine lathe, except for the difference in the path of the tool, the tool having a circular instead of a straight path.

It is evident that the worm shape is dependent on the particular wheel with which it is to run, and Hindley worms are not interchangeable with any other but an exact duplicate. That is, a worm cut for a Hindley gear of 50 teeth cannot be used successfully with a wheel of 70 teeth, although the pitch of the teeth is exactly the same. In the ordinary type of worm gearing, one worm may be made to run with any number of diameters of wheels of the same pitch, and hobbed with the same hob.

In action the two styles of worm-gear differ greatly, and both diverge widely in action from the case of a plain nut and screw, which may be taken to represent a worm and worm-gear, the latter of infinite diameter and with an angle of embrace of 360 degrees. In studying the action between the thread and teeth of the ordinary type of worm-gear, we must understand odontics, rolling contacts and the theory of tooth gearing, in general, in order to understand the action of the ordinary worm-gear. But, in studying the action of the Hindley type, we are concerned with no such theories, as the action is purely sliding and devoid of rolling contact. In the ordinary worm we have an axial pitch which is constant from top to root of the thread, while in the Hindley worm we have a section in which the pitch of the thread varies from top to bottom.

The interference in the ordinary type of worm-gear is absent from the Hindley type, and the consequent undercutting and weakening of the teeth, therefore, is a feature with which the designer of the Hindley worm gearing does not have to contend. For this reason we are not limited in the length of teeth, by interference, as in the ordinary case. This fact permits a wide latitude in the choice of tooth shapes and proportions. In most examples we will find that the depth of thread is much greater in proportion to the thickness than in the ordinary worm-gear, in which the height is limited by reason of the interference at the top and root of the teeth.

Nature of Contact of Hindley Worm Gearing

The general idea of the Hindley worm gearing is that there is surface contact between the worm and gear, and that the contact is generally over the whole number of teeth in mesh. If such were the actual conditions, the Hindley type would surely be an ideal mech-

anism for high velocity ratios, but that such is not the fact is the purpose of this treatise to point out. That the contact is of a superior nature we will not deny, nor that it is much nearer a surface contact than exists in the ordinary worm gear. As a means of comparison, Figs. 25 and 26 are shown. Fig. 25 shows an axial section taken through the worm and gear of the ordinary type, while Fig. 26 shows a similar section through the Hindley worm and gear. The "airy"

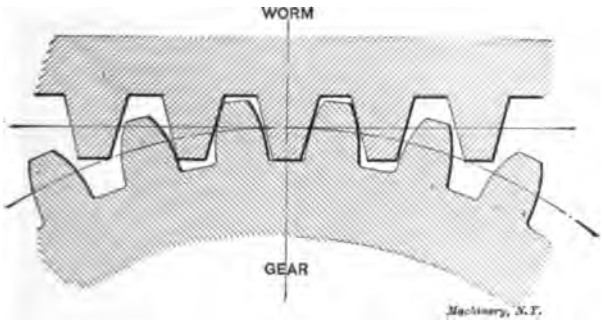

Fig. 25. Section of Common Worm and Worm-wheel on Middle Plane

appearance of Fig. 25 as compared with Fig. 26, indicates a vast difference in the nature of contact, and gives the advantage to the Hindley type, wherein is the origin of certain false ideas in favor of the latter. These illustrations also show peculiar differences in the action of the two types. The absence of rolling action in Fig. 26 is the most prominent, and it shows the similarity between this type of gear and a screw and nut.

From an inspection of Fig. 26 we may feel sure that the contact on

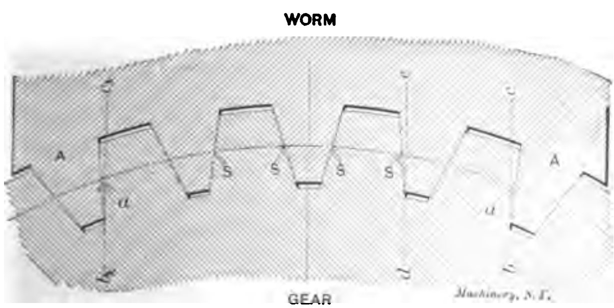

Fig. 26. Section of Hindley Worm and Gear on Middle Plane

the axial plane is as shown, but as to the nature of contact in a plane either side of the middle plane we are in the dark so far as the drawing illustrates. Mr. George P. Grant has this to say concerning the contact of the Hindley worm and gear: "It is commonly but erroneously stated that the worm (Hindley) fits and fills its gear on the axial section. . . . It has even been stated that the contact is between surfaces, the worm filling the whole gear tooth. . . . It is

also certain that it (the contact) is on the normal and not on the axial section, and that the Hindley hob will not cut a tooth that will fill any section of it. The contact may be linear on some line of no great length, but it is probably a point contact on the normal section."

It is not clear what reason Mr. Grant had for saying that the contact is normal instead of axial; but there is every reason to believe that the contact is on the axial section since it is on this section that the teeth of the hob have a common pitch. The teeth have not a common pitch on any section at an angle with the axial section. For what reason would one expect to find contact on the normal section in this case any more than in the case of the ordinary worm? Since both styles of worm-wheels are hobbed with a revolving hob which lies in a plane perpendicular to the axis of the worm-wheel, the contact could hardly be on a normal section.

Prof. MacCord states that he considers the contact to be line contact on the axial section, and he gives directions for obtaining the exact nature of the contact and also the thread and tooth sections. These directions, on account of the complicated nature of the method,

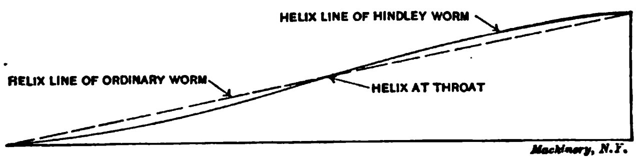

Fig. 27. Development of Ordinary Worm and Hindley Worm Spirals on a Plane

are hard to follow. Much, however, can be found out by simple methods. In what follows, describing these simpler methods, the results, of course, are of an approximate order, but they nevertheless give a means of comparison and a material basis for the line of argument.

The Ideal Case Considered

It is assumed that we are examining an ideal Hindley gear in which the worm and wheel are theoretically correct in shape and that the surfaces are perfectly smooth and inelastic. From the nature of the worm, the helix angle varies from mid-section to the ends, decreasing as the thread approaches the ends of the worm. The thread is spiral as well as helical. This change in the thread angle is caused by the increase in diameter at the ends of the worm and by the fact that the axial pitch of the thread decreases as it reaches the ends. The decrease in axial pitch is due, of course, to the circular path of the threading tool. If we take a development on a flat surface of a line scribed in the spiral path on the worm blank, as shown in Fig. 27, the change in the angle becomes noticeable.

In the operation of forming the teeth of the gear, the blank is rotated, each portion of the hob working the tooth into shape so that it will pass the corresponding portion of the worm thread without interference, permitting a smooth transmission of motion. If each

portion has a different shape or is placed in a different relation, the shape of a gear tooth will be a compromise between the extremes, and this is what is actually the result, as we shall see later.

The progressive steps of the process are shown in Fig. 28; the successive positions of one tooth are shown, beginning at the left and ending at the right-hand position where each tooth is given its final shape. The nature of the process is shown in Fig. 29, the shaded portions representing the gear teeth. Here we have a representation of the contact of the thread and teeth; it shows that surface contact is impossible on any but the heavily shaded portions of the teeth, it being confined to the mid-section and the extreme end sections of the worm. Line contact is obtained throughout the length of the worm on the axial plane. This figure also shows that no advantage is gained in surface contact by making the worm of greater length. The location of the contacts are shown in Fig. 29, at *a*, *s*, *s*, *s*, *s*, *a*, but it

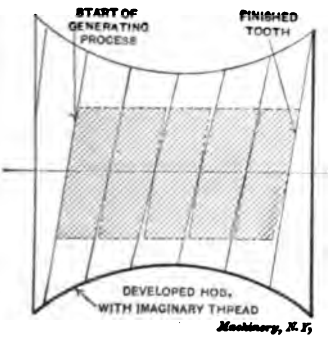

Fig. 28. Successive Steps in Shaping the Hindley Worm

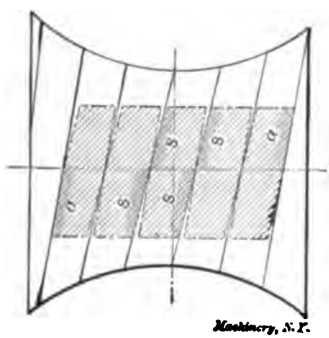

Fig. 29. Surfaces of Contact of the Hindley Worm

must be remembered that they lie on opposite sides of the cutting plane. From this it is apparent that the worm does not entirely fill the space between the teeth of the gear and that the contact is not wholly a surface contact.

Let us investigate still further and see whether the conditions are not modified by other irregularities: Fig. 30 is drawn to represent a worm and gear of the Hindley type, in mesh, the teeth of which have no depth. As before mentioned, the peculiarity of this type of worm is its hour-glass shape. The hob and worm may be treated as identical in form. In the process of generation, the tooth has a pitch line curvature that changes with corresponding positions in relation to the thread portion acting upon it. The tooth must necessarily be modified from what it should be for any particular location in its contact with the worm thread. It is quite clearly shown that if the tooth is to fill the worm thread or *vice versa*, it must be formed in strict accordance with the thread at that particular point. Thus if at *j* the tooth fills the thread, that tooth must be formed by the thread at that point, while the tooth at *k* must be formed by the thread at *k*. Now, since each tooth must pass from *k* to *j*, its form must be such

that it will do so without interference. It is evident that the radial section of the gear at *k* must be the same as at *j*. Since the worm is largest in diameter at *k*, the curvature of the tooth on the radial section is dependent on the thread at that point. The curvature of the tooth at *k* evidently is that of an ellipse whose major axis is $A A_1$. Now, since the thread is made with angular sides, the hob could hardly act on the teeth of the gear the same at all points from *k* to *j* except on the axial plane where the relative shape of the hob thread is the same for any position along the line of action (see Fig. 26). This is evident from Fig. 30 at *E*, which point only touches at the mid-section of the worm. Therefore we still have the line contact from top to bottom of teeth on the axial plane, but the construction, Fig.

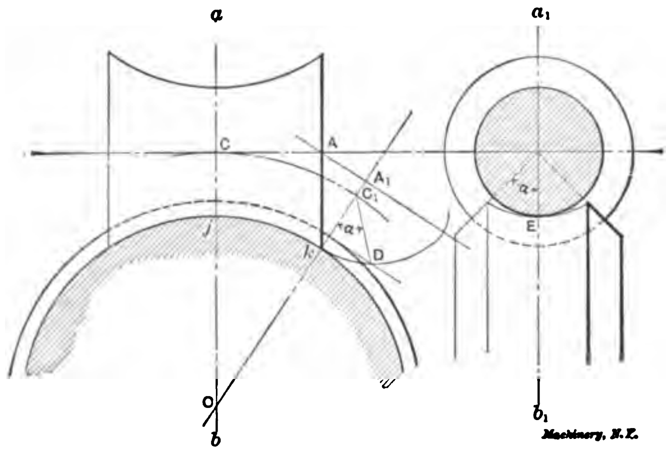

Fig. 30. Effect of Hour-glass Shape on Worm-wheel Contact

30, shows that the surface contact *s*, *s*, *s*, *s*, Figs. 26 and 29, does not actually exist, but that the surface contact at the ends of the worm remains undisturbed.

From the above we may safely conclude that the hob at *j* has but little effect on the actual shape of the tooth, and that its influence increases until *k* is reached. Fig. 30 also shows a good reason why the contact may be considered axial instead of normal, by the mere fact of the differences in curvature of worm and wheel at any point other than *k*. In practice the contact may appear to be surface contact, but this, no doubt, is due to the influence of the lubricating oil and the fact that materials of construction are distorted to some extent in form when subjected to pressure. This distortion permits the worm thread to imbed itself into the worm-wheel teeth, somewhat broadening the contact for the time being. The conditions as stated in the above discussion would be met in the case of a hardened worm and gear with surfaces finished by lapping. In practice the worm and gear are ground together, sand and water being used as the abrasive. This grinding wears down the roughness of the surfaces and tends to correct irregularities in form that develop in the hobbing process.

Objections to the Hindley Gear

The objections to the Hindley type of worm-gear are many and are widely known. It must be set up accurately, the alignment being made perfect. End play is a feature that must be avoided, as any longitudinal displacement of the worm will cause the gear to cut. These peculiarities are the greatest drawbacks to the use of this gear, and because of them the writer believes that it will not come into common use, at least not so common as the worm drive of the ordinary type. This opinion is strengthened by the fact that we have become so much more familiar with the latter type as to be able to design and construct drives that work satisfactorily in every respect.

Modifications of Hindley Worm-Gear Practice

Some modifications have been made in the process of manufacturing the Hindley worm-gear. One that is probably of first importance is that known as the "second cut," this practice being generally cred-

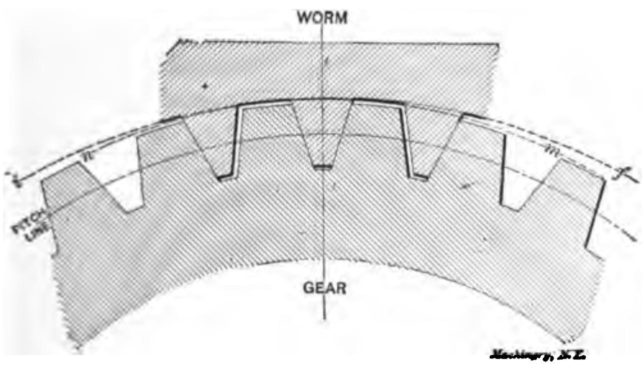

Fig. 31. Effect of the "Second Cut" on Contact

ited to Mr. Albro of Philadelphia, but the credit for it is in dispute. The effect of the second cut is indicated in Fig. 31. From this illustration one would say that the object of the second cut is to remove the points of contact. Whether this is the reason or not, it is a fact that it does remove considerable of the contact from all but the mid-section of the worm. This second cut is made by enlarging the diameter of the circle in which the threading tool travels when cutting the worm. It is said to have advantages that add to the wearing quality of the drive, but just what these advantages are is not apparent, and since the process is considered more or less a trade secret, it is difficult to obtain authentic reasons for its use.

The limiting length of the worm is dependent on the shape of the thread. In Fig. 31 the worm is shown with three teeth in mesh, while Fig. 26 shows five. Fig. 26 shows a case that would be impossible in practice on account of the undercut teeth *A* which lock the worm in mesh. The side of the thread must fall inside the line *b c* to permit the worm and gear to be assembled.

Conclusions Regarding the Hindley Worm and Gear

The following are the author's conclusions, derived from the investigation regarding the Hindley type of gear:

1. The contact is purely sliding contact.
2. The nature of the contact is linear, closely resembling surface contact.
3. Linear contact extends from the top to the root of the tooth.
4. The contact is on the axial section.
5. The thread section fills the tooth space on the axial section only.
6. The mid-portion of the hob has little or no effect in shaping the teeth of the gear.
7. Surface contact exists on opposite sides of the axial plane at the end of the worm thread and is intermittent in nature, because the end of the thread passes out of contact with the tooth in the revolving of the worm. This contact is on a plane normal with the thread angle.

In practice it is usual to allow considerable back-lash between the thread and the tooth of the worm and gear. This play tends to counteract bad workmanship, either in construction or erection.

CHAPTER VI

THE DESIGN OF SELF-LOCKING WORM-GEARS*

The old opinion that the friction and wear of worm-gears are necessarily very great, and that the efficiency is necessarily very low, making worm gearing an unmechanical contrivance, is not as frequently met with now as formerly. In Unwin's Machine Design it is stated that in well fitted worm gearing, of speed ratios not exceeding 60 or 80 to 1, motion will be transmitted backwards from the wheel to the worm. In Prof. Forrest R. Jones' work on machine design may be found tabulated the results of many examples from practice, some of which show an efficiency as high as 74 per cent before abrasion began, the most notable example being that of a worm running at a surface speed of 306 feet per minute under a load of 5,558 pounds, and showing an efficiency of 67 per cent, with no abrasion. The tables in Prof. Jones' work show that under light loads very high surface or rubbing speeds are allowable, running as high as 800 feet per minute. It has also been pointed out that an increase in the thread angle, in general, increases the efficiency.

There is, however, an important function of worm gearing which is not, as a rule, brought out adequately by writers on worm gearing, and which in certain classes of machinery is of the first importance; often, indeed, becoming the determining factor in deciding upon the choice of a worm-gear as the power transmitter. It is the property a worm-gear possesses, under certain conditions dependent upon its design, of being self-locking, and preventing motion backwards.

An instance where this property becomes of prime importance and accounts for the use of the worm-gear, is in crane work, where the winding drum is driven by a worm-gear so designed that, when the power is shut off, the gear will not run down or backwards under the impulse of the load, but will be self-locking, holding the load at any point.

Fig. 32 shows a single thread worm in mesh with the worm-wheel, a being the angle of the worm thread with the axis of the worm-wheel, and in order that the system may be self-locking, that is, that the worm-wheel may be unable to run the worm, the tangent of the angle a must be less than the coefficient of friction between the teeth of the worm and wheel, or as

$$\tan a = \frac{p}{\pi d}, \text{ so } \frac{p}{\pi d} < f \qquad (1)$$

in which p = the pitch; d = the pitch diameter of the worm; and f = the coefficient of friction between the worm and wheel. It is neces-

* MACHINERY, December, 1902.

sary to assume a value for f, which, if the condition of determining the use of the worm-gear is its self-locking property, should be assumed conservatively low. Unwin states under the authority of Prof. Briggs, that a well-fitted worm-gear will exhibit a lower coefficient of friction than any other kind of running machinery. Prof. Jones gives a series of values for the coefficient of friction of screw gears, one of which is a pinion of 4 inches pitch diameter, the average value being $f = 0.05$, corresponding to a rubbing velocity of 250 feet per minute. Mr. Halsey assumes $f = 0.05$, and Mr. Wilfred Lewis says that when the worm-gear is worked up to the limit of its safe strength, a rubbing velocity greater than 200 to 300 feet per minute will prove bad practice. It is in heavy machinery where worm-gears are mostly used as self-locking transmission elements, and here they are usually worked up to the safe strength of the wheel; hence it is fair to assume $f = 0.05$

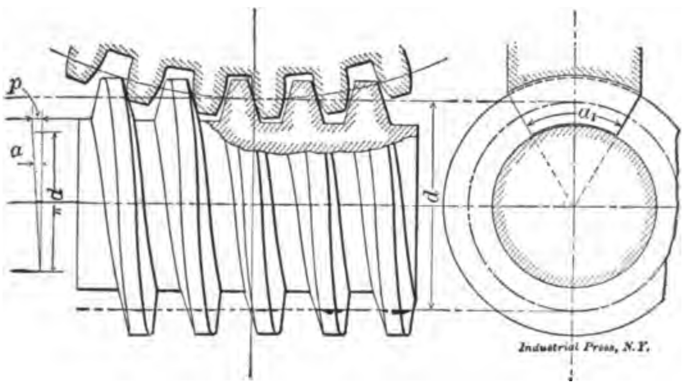

Fig. 32

when designing a self-locking worm-gear, and to limit the rubbing velocity to 200 feet per minute, and we have for the limiting value of p at which the system will be self-locking:

$$p = 0.05\,\pi\,d = 0.157\,d \qquad (2)$$

The sliding velocity in feet per minute at the pitch line is expressed by

$$V = \frac{\pi\,d\,n}{12} = 0.262\,d\,n\,. \qquad (3)$$

where d = the pitch diameter of the worm, and n = the number of revolutions per minute of the worm.

Under the above assumption, that for continuous service and heavy pressures the sliding velocity should not be more than 200 feet per minute, we have as the limiting value of d to avoid all cutting:

$$d = \frac{200}{0.262\,n}.$$

The exact nature of the surface of contact between a worm and wheel is involved in doubt; many claim it is only a point; it certainly is not large, and consequently a wide face for the wheel is not needed

If the angle a_1 is made 60 degrees, it will make the face right for any ordinary worm of 4 to 6 inches diameter.

There is in all worm gearing a very heavy end thrust on the worm-shaft, and also an outward force normal to the worm-axis, each of which must be suitably provided for in the design of the shaft and bearings. The end thrust may be taken by bronze washers slipped into the bearings at the end of the shaft, which may be removed when worn and replaced with new ones. Shoulders may be provided on the shaft, between which and the bearings bronze collars may be placed, these being split to enable new ones to be easily and quickly placed in position when the old ones become worn. Roller thrust bearings are very often applied to worms, and these as well as the bronze washers may be supplied with adjusting set-screws to take up the wear, instead of renewing the washers.

Analysis of the Forces

In Fig. 33 let P = the tangential force at the pitch line of the worm, d = the pitch diameter of the worm, Q = the tangential force at the

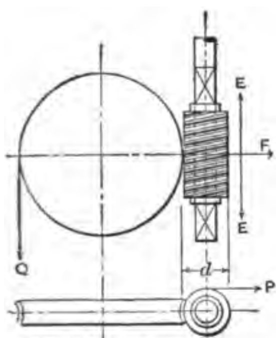

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Fig. 33

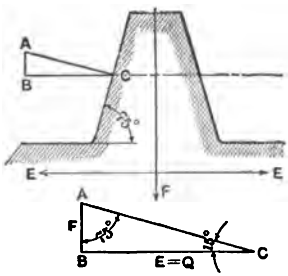


Fig. 34

pitch line of the worm-wheel, E = the end thrust of the worm-shaft, F = the force on the worm-shaft normal to the worm-axis; then, friction being neglected,

$$Q = \frac{P \pi d}{p} \qquad (4)$$

In Fig. 34, draw line BC parallel to the axis, or coinciding with the pitch line, of the worm; let this line represent the force $E = Q$; draw AB normal to this line; it will then also be normal to the axis of the worm; then, when measured to the same scale to which BC is drawn, $AB = F$; if the angle CAB is 75 degrees, we have:

$$\frac{F}{Q} = \tan\ 15\ \text{deg.} \qquad (5)$$

$$F = 0.268\ Q \qquad (6)$$

Taking friction into consideration, the force P_1, tangential to the pitch

line of the worm, which it is necessary to employ in order to produce a force Q tangential to the pitch line of the wheel, is given by Weisbach as

$$P_1 = Q \frac{h + f}{1 - hf} \tag{7}$$

in which

$$h = \frac{p}{\pi d}$$

The efficiency of the worm and wheel is then,

$$\frac{P}{P_1} = e \tag{8}$$

Example: A single thread worm of 1-inch pitch, running 80 revolutions per minute, is to transmit to a worm-wheel a tangential force $Q = 5{,}000$ pounds, and is to be self-locking.

From (3)

$$d < \frac{200}{0.262 \times 80},$$

or d may be as large as 9.5 inches before abrasion need be feared.

From (2)

$$p < 0.157\ d;\ \text{assume}\ p = 0.125\ d,$$

then, as $p = 1$ inch, $d = 8$ inches, or the worm will require to be 8 inches pitch diameter in order that the angularity of the thread may be small enough to make the system self-locking. It will be seen that the required diameter will be increased as the value of f is decreased, and in case the required diameter of the worm proves too great for practice, and the pitch cannot be reduced on account of considerations of strength, some outside aid, such as a brake or friction disk applied to the worm-shaft, will have to be adopted.

From (7)

$$\text{as}\ h = \frac{p}{\pi d} = \frac{1}{3.14 \times 8} = 0.04,\ \text{we have}$$

$$P_1 = 5{,}000 \frac{0.04 + 0.05}{1 - (0.04 \times 0.05)} = 451\ \text{pounds.}$$

From (4)

$$5{,}000 = \frac{3.14 \times 8 \times P}{1},\ \text{or}\ P = 199\ \text{pounds.}$$

From (8)

$$\frac{P}{P_1} = \frac{199}{451} = 44\ \text{per cent for the efficiency of the worm-gear.}$$

The formulas may, by starting with those for the efficiency, be used to determine the pitch diameter which will give the proper thread angle for any given pitch and degree of efficiency.

It is clear from the foregoing, that a worm-gear of large pitch will require a pitch diameter of the worm altogether too large for practice, if it is to be self-locking, and that the system as usually designed may be expected to run backwards. To prevent this, a friction disk may

be placed in the bearing which receives the thrust of the worm-shaft when the system is running backwards, and the diameter of the disk so proportioned as to just hold the worm-shaft stationary under the impulse of the worm-wheel.

The foregoing discussion neglects the effect of the thrust of the worm-shaft in its bearings, the frictional resistance of which must be added to that of the teeth to obtain the actual conditions of a self-locking system. This frictional resistance depends upon the values of the end thrust E and the normal force F already found, and the diameter and form of the bearing. In nearly all cases of worm gearing the mounting of the worm upon the shaft will be covered by one

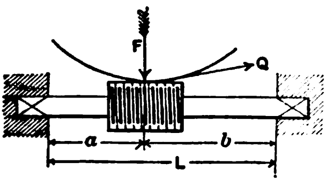


Fig. 35

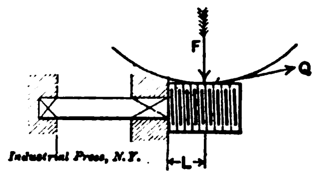


Fig. 36

of three cases, either unsymmetrically between the bearings, symmetrically between the bearings, or over-hung.

In Case 1, Fig. 35, the bending moment upon the worm shaft is,

$$M = \frac{F\,a\,b}{L} = \frac{0.259\,Q\,a\,b}{L} \qquad (9)$$

In Case 2, same as Case 1, except that the worm is central between the bearings, and

$$a = b = \frac{L}{2}$$

the bending moment upon the worm-shaft is,

$$M = \frac{0.259\,Q\,L}{4} = 0.0647\,Q\,L \qquad (10)$$

In Case 3, Fig. 36, the bending moment upon the worm-shaft is,

$$M = F\,L = 0.259\,Q\,L \qquad (11)$$

In each of the above cases the shaft is subjected to a combined twisting and bending strain, the twisting moment being the same in each case, $T = P\,R$, which is, however, so small as to be negligible in what follows.

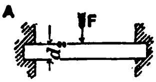
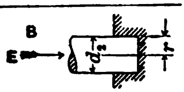
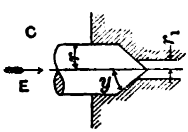
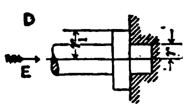
In the following table the first column shows the several styles of journals most commonly used for worm-shafts, the second column gives the moment of friction for each under a load in the direction of the arrow, the third column gives the coefficient of friction assumed, and the fourth column gives the tangential force P_2 at the pitch line of the worm, resulting from the resistance of friction in the journals, and found by dividing the moment of friction in Column 2 by the pitch radius of the worm.

There are always acting upon the worm-shaft the two forces F and

E; consequently to get the resultant retarding force tangential to the pitch line of the worm, we must take the sum of the resultants due to the frictional resistance of each force separately. Referring to the table, we will, for each worm-shaft, find the conditions shown at A, in addition to the conditions shown either at B, C or D, as the case may be, and the total resultant force P_2 at the worm pitch line will be the sum of the quantities given in Column 4 opposite the particular cases.

These frictional resistances developed by the journals act in a direction helpful to the self-locking property of the worm, and enable the designer to use a larger thread angle for a given diameter of worm,

TABLE GIVING MOMENT OF FRICTION WITH VARIOUS TYPES OF BEARINGS

Style of Journal.	Moment of Friction.	f	$\frac{\text{Moment of Friction}}{R}$
A 	$\frac{f F d_1}{2}$	.05	$\frac{.04 P d_1}{p}$
B 	$\frac{2 f E r}{3}$	.05	$\frac{.2 P r}{p} = \frac{.1 P d_1}{p}$
C 	$\frac{2 f E (r^3 - r_1^3)}{3 r \sin. y}$	.05	$\frac{.2 P (r^3 - r_1^3)}{p r \sin. y}$
D 	$\frac{2 f E (r_1^3 - r^3)}{3 (r_1^2 - r^2)}$	.05	$\frac{.2 P (r_1^3 - r^3)}{r_1^2 - r^2}$

or a smaller diameter of worm for a given thread angle, thus keeping within the limits of good practice, and increasing the efficiency of the system for the forward movement.

Having determined the force P_2 tangential to the worm pitch line, resulting from the frictional moment at the journals, the angle of repose for this force acting with the force Q, as shown in Fig. 37, is given by the equation,

$$\tan x = \frac{P_2}{Q}.$$

The thread angle found previous to the consideration of the effect of the journal friction may now be increased by the angle x, making the thread angle $a + x$. This may be accomplished either by increasing the thread angle, increasing the pitch, or decreasing the pitch diameter.

Consider. now, that in the foregoing example, the worm-shaft is of

the form in Case 2, the worm being central between the bearings, and the distance between bearings being 36 inches.

Then, from (5) we have,

$$F = 0.259 \times 5{,}000 = 1{,}295 \text{ pounds,}$$

and from (6)

$$E = 0.966 \times 5{,}000 = 4{,}830 \text{ pounds.}$$

From (10)

$$M = \frac{0.259 \times 5{,}000 \times 36}{4} = 11{,}655 \text{ inch-pounds.}$$

Assuming $s = 10{,}000$ pounds per square inch for the allowable fiber stress in the worm-shaft, we have

$$M = \frac{\pi}{32} d_2^3 s \text{ or } d_2 = 2.28 \text{ inches.}$$

From the table, Case A,

$$P_1 = \frac{0.04 \times 199 \times 2.28}{1} = 18.15 \text{ pounds.}$$

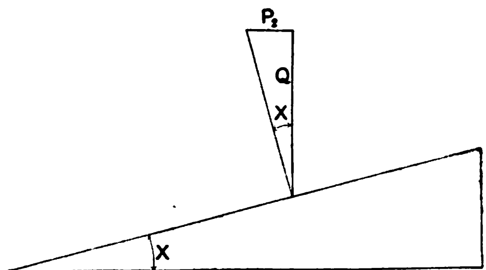


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Fig. 37

From the table, Case B,

$$P_1 = \frac{0.1 \times 199 \times 2.28}{1} = 45.37 \text{ pounds.}$$

Then

$$P_2 = 18.15 + 45.37 = 63.52 \text{ pounds.}$$

$$\tan x = \frac{63.52}{5{,}000} = 0.0127$$

$$x = 0 \text{ deg. } 44 \text{ min.}$$

From (1)

$$\tan a = \frac{1}{3.14 \times 8} = 0.04$$

$$a = 2 \text{ deg. } 17 \text{ min.}$$

Then

$$a + x = 3 \text{ deg. } 1 \text{ min.}$$

$$\tan 3 \text{ deg. } 1 \text{ min.} = 0.053$$

$$\frac{p}{\pi d} = 0.053 = h, \text{ and } d = 6 \text{ inches, approx.}$$

If, now, we substitute these new values of h and d in equations (7) and (4), we continue as follows:

From (7)

$$P_1 = 5{,}000\,\frac{0.053 + 0.05}{1 - (0.053 \times 0.05)} = 516 \text{ pounds.}$$

From (4)

$$5{,}000 = \frac{P \times 3.14 \times 6}{1}, \text{ or } P = 265 \text{ pounds.}$$

From (8)

$$\frac{P}{P_1} = \frac{265}{516} = 51 \text{ per cent efficiency for the worm-gear.}$$

The total efficiency of the system, taking account of the journal friction, will be

$$\frac{P}{P_1 + P_2} = \frac{265}{516 + 63.52} = 46 \text{ per cent.}$$

It thus becomes clear that while the efficiency of the worm threads and wheel teeth has been increased above 50 per cent, the efficiency of the whole system, including the journals, is below 50 per cent, and the system retains its self-locking property. It is evident that when running forward, the end thrust E upon the worm-shaft will be upon the opposite end from that when running backward, and on this account a system may be designed to have a high efficiency on the forward movement and still preserve its self-locking property.

If both the journals have roller bearings, and the end taking the thrust on the forward movement has a ball bearing, while the opposite end be made like Case C or D in the table, properly proportioned, the worm may be designed to show a very high efficiency on the forward movement, while the frictional resistance of the step bearing on the opposite end will cause the system to be self-locking by reason of the energy absorbed at the step bearing.

The formulas may be put into more convenient form for this purpose, as follows:

The designer will have, to start with, a knowledge of the force Q required at the worm-wheel, the force P_1 at the pitch line of the worm, developed from the source of power, the pitch required for the worm-wheel, and the efficiency e for which he wishes to design the system. We then have,

$$\frac{P}{P_1} = e, \text{ and } P = P_1\,e.$$

Substituting this value for P in equation (4) and solving for d, we have

$$d = \frac{p\,Q}{3.14\,P_1\,e}$$

for the worm, neglecting the journals, when the journals and thrust bearings are roller and ball bearings, respectively, and

$$d = \frac{p\,Q}{3.14\,(P_1 - P_2)\,e}$$

when the journals and thrust bearings are considered.

The worm being thus designed for the given efficiency on the forward movement, it remains to determine such proportions of the step bearing for the backward movement as will present enough frictional resistance to render the system self-locking. Let e_1 = the efficiency when the journals and thrust are considered, then

$$\frac{P}{P_1 + P_2} = e_1 \text{ or } P = e_1 (P_1 + P_2)$$

and substituting the value of P found above

$$e P_1 = e_1 P_1 + e_1 P_2$$

and

$$P_2 = \frac{P_1 (e - e_1)}{e_1}.$$

By equating this force P_2 to the proper quantity from Column 4 in the table of journal resistances, the proportions required of the journal or step bearing may be determined.

Theoretical Efficiency of Worm Gearing

The following table gives the theoretical efficiency of worm gearing for a number of different coefficients of friction. Practical experiments carried out by the Oerlikon Company, Oerlikon by Zürich, Switzerland, agree closely with the results from theoretical calculations given in the table. These experiments indicate that the efficiency increases

TABLE GIVING THEORETICAL EFFICIENCY OF WORM GEARING.

Coefficient of Friction.	Angle of Inclination.								
	5 deg.	10 deg.	15 deg	20 deg.	25 deg.	30 deg.	35 deg	40 deg	45 deg
0.01	89.7	94.5	96.1	97.0	97.4	97.7	97.9	98.0	98.0
0.02	81.3	89.5	92.6	94.1	95.0	95.5	95.9	96.0	96.1
0.03	74.8	85.0	89.2	91.4	92.7	93.4	93.9	94.1	94.2
0.04	68.4	80.9	86.1	88.8	90.4	91.4	92.0	92.2	92.3
0.05	63.4	77.2	83 1	86.3	88.2	89.4	90.1	90.4	90.5
0.06	59.0	73.8	80.4	84 0	86.1	87.5	88.2	88.6	88.7
0.07	55.2	70 7	77 8	81.7	84.1	85 6	86.4	86.9	86 9
0.08	51.9	67.8	75.4	79 6	82 2	83.8	84.7	85.2	85.2
0.09	48.9	65.2	73.1	77.6	80.3	82.0	83.0	83.5	83.5
0.10	46.8	62.7	70.9	75.6	78.5	80 3	81.4	81.9	81.8

with the angle of inclination, up to a certain point. They also show that for larger angles of inclination than 25 degrees to 30 degrees the efficiency increases very little, especially if the coefficient of friction is small, and this fact is of importance in practice, because, for reasons of gear ratio and conditions of a constructive nature, an angle greater than 30 degrees cannot be employed. The coefficient of friction increases with the load and diminishes to a certain extent with increase of speed. Besides the friction between the worm and the wheel teeth, there is also the friction of the spindle bearings and the ball bearings for taking the axial thrust. To obtain the best results,

there must be very careful choice of dimensions of teeth, of the stress between them, and the angle of inclination. To show what can be done, the following are the results of a test with an Oerlikon worm-gear for a colliery winding engine: The motor gave 30 brake horse-power to 40 brake horse-power at 780 revolutions. The normal load was 25 brake horse-power, but at starting it could develop 40 brake horse-power. The worm-gear ratio was 13.6 to 1, the helicoidal bronze wheel having 68 teeth on a pitch circle of 7.283 inches, and the worm 5 threads. The power required at no load for the whole mechanism was 520 watts, corresponding to 2.8 per cent of the normal. The efficiency at one-third normal load gave 90 per cent, at full load 94½, and at 50 per cent overload 93 per cent. The efficiency of the *worm and wheel* alone is higher, and knowing the no-load power, is calculated to be 97½ per cent. According to the table given, of theoretical efficiencies, this gives the coefficient of friction as 0.01. To obtain a reduction of 13.6 to 1 with spur gears would have necessitated two pinions and two wheels with their spindles and bearings, and if the bearing friction was taken into consideration, the efficiency of such gearing would certainly not have reached the above-mentioned figure of 94½ per cent at full load. These figures, of course, seem very high for the efficiency of worm gearing. They were published in MACHINERY, December, 1903, having been obtained from a reliable source, and were never challenged.

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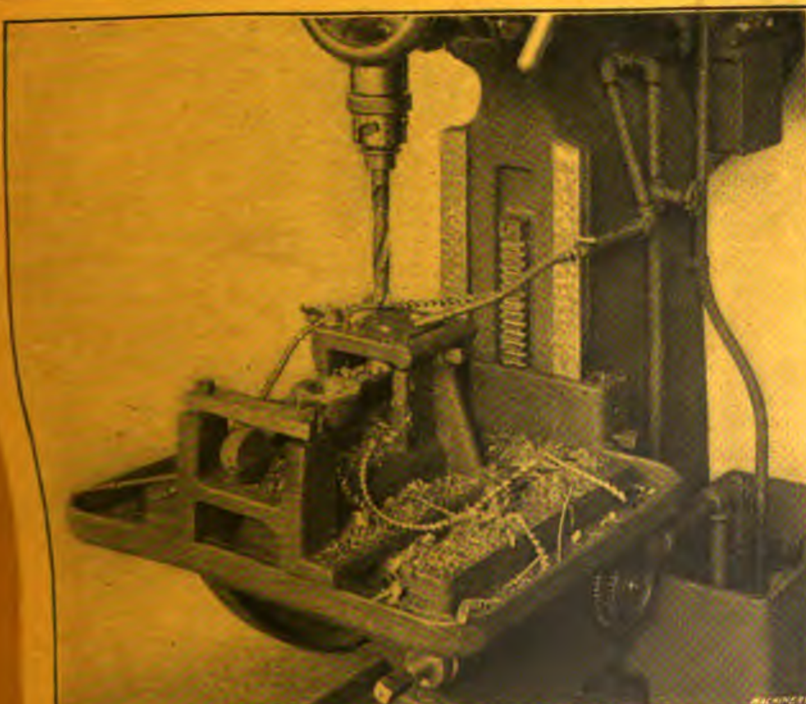
DRILL JIGS

PRINCIPLES OF DESIGN—EXAMPLES FROM PRACTICE DIMENSIONS OF JIG BUSHINGS

THIRD EDITION



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NUMBER 3

DRILL JIGS

THIRD EDITION

CONTENTS

CHAPTER I

ELEMENTARY PRINCIPLES OF DRILL JIGS*

The reasons for the use of jigs may be summed up under three heads, the order in which they are stated representing fairly well the frequency of occurrence, though not necessarily the importance, of these reasons: First, reduction of cost; second, duplication; third, accuracy.

Let us first consider the question of cost. As no article can, as a rule, be sold in open competition with similar articles unless its cost is somewhat proportionate to the quality of its competitors, commercial considerations demand that the cost be kept as low as possible, while the quality be kept as high as possible; and jigs are one of the chief agents of this in metal work. When a jig is considered, the first thing to be settled is whether it can be made to pay, and if so, how much. The answer to this often involves very many other questions, but can generally, if not always, be resolved into computations based upon the number of pieces to be made, and the probable cost of labor per piece when made with and without a jig, and the cost of the jig, including maintenance. Also the fact that often a much less valuable machine, or one less busy, can be used with the jig, may be an important consideration. If no other factor than cost of production is involved, and it is found that the total cost of the jigged work will come very near that of the lot of articles when made without a jig, and no further order is in sight, it is pretty safe to abandon the jig idea; for it is apt to partake very much of the nature of an experiment, and the odds should be decidedly favorable to warrant the risk.

The second reason—the duplication of pieces—has a somewhat different foundation— though cost enters here also, as will be seen later. Suppose the part to be made is subject to wear or breakage, as in agricultural and textile machinery, guns, bicycles, etc. We know, for instance, the strong disinclination anyone has for buying a wheel, the makers of which have gone out of business. It is at once recognized that repair parts cannot be bought from stock dealers, but must be made at excessive cost and delay. So we have before us the importance to manufacturers that the buying public shall have confidence in the interchangeability of parts in order that sales may be made at all upon the open market. It is a fact that where this reason holds good, there is also the reason that costs will be lessened, because production of large numbers of parts is taken for granted. And in considering whether or not a jig shall be made, this combination of reasons militates strongly for the jig. There is also another equally important reason for jigs, based on costs and interchangeability—it is that, in

* MACHINERY, October, 1902; November, 1906; December, 1906, and January, 1907.

fitting and assembling, those parts which are exactly alike require a minimum amount of labor when putting in place. This, perhaps, one may, without danger of exaggeration, say is in most cases in the machine building business the chief consideration.

In the third place, accuracy is often attained only by the use of jigs. There are certain classes of work which could not be finished at all within the limits of accuracy demanded, if jigs of some sort were not used.

It will therefore be seen that the determination of whether a jig shall be made may rest upon a number of questions which often demand great care and practical experience to solve in the way best meeting the requirements of the case.

Drill Jigs

Drill jigs are used for drilling holes which must be accurately located, both in relation to each other and to certain working surfaces and points; the location of the holes is governed by holes in the jig through which the drill passes. The drill must fit the hole in the jig to insure accuracy of location. When the jig is to be used in drilling many holes, the steel around the holes is hardened to prevent wear. If extreme accuracy is essential, or if the jig is to be used as a permanent equipment, bushings, made of steel and hardened, are used to guide the drills.

General Considerations in Designing Jigs

The design of a jig should depend altogether on the character of the work to be done, the number of pieces to be drilled, and the degree of accuracy ncessary in order that pieces drilled may answer the purpose for which they are intended. When jigs are to be turned over and moved around on the drill press table they should be designed to insure case and comfort to the operator when handling, and should be made as light as is consistent with the strength and stiffness necessary. Yet, we should never attempt to save a few ounces of iron, and thereby render the jig unfit for the purpose we intend to use it for. The designer should see that the jig is planned so that work may be easily and quickly placed in and taken out, and that it can be easily and accurately located in order to prevent eventual mistakes. As it is necessary to fasten work in the jig in order that it may maintain its correct position, fastening devices are used; these should allow rapid manipulation, and yet hold the work securely to prevent a change of location. Yet, while it is necessary to hold work securely, we should not use fastening devices which spring the work, or the holes will be not only improperly located, but they will not be true with the working surfaces or with each other. When finishing the surfaces of drill jigs and similar devices used in machine shops, the character of the finish depends entirely on the custom in the shop, for while in some shops it is customary to finish these tools very nicely, removing every scratch, and producing highly finished surfaces, in other shops it is not required, neither is it allowed, as it is considered a waste of time and an unnecessary item of cost.

Limits of Accuracy

When making drill jigs we must discriminate between measurements that must be *exact*, and those not requiring extreme accuracy; it is not considered good practice, and it shows poor judgment, to spend the amount of time necessary to locate a hole within a limit of variation of 0.001 inch or even closer, if a variation of 1/64 inch is insignificant. But if the holes must be located *exact* as to measurements, it is necessary to work as accurately as possible, and time cannot be considered a factor, provided a man improves every minute. Yet the fact that extreme accuracy must be observed does not warrant a jigmaker *wasting time*.

Before starting to work on tools of this character, the workman should first carefully look over his drawing, making himself thoroughly familiar with the construction, and making sure that the measurements given are, seemingly, correct; if in doubt about anything, consult the foreman, or the draftsman—according to the custom in the shop—in order that every detail may be thoroughly understood, or that any mistake made in the drawing may be rectified.

Many times one draftsman is puzzled to understand a drawing made by an equally good man, especially so if the work is foreign to him; and a shop man, who may not be very well versed in reading drawings—yet be an excellent workman—may easily get puzzled when he attempts to read a drawing of work he is not familiar with. Inquiries and proper explanations are therefore in place, and there should be no hesitation about asking questions, nor any reluctance about replying to them.

Provisions for Chips and Burrs

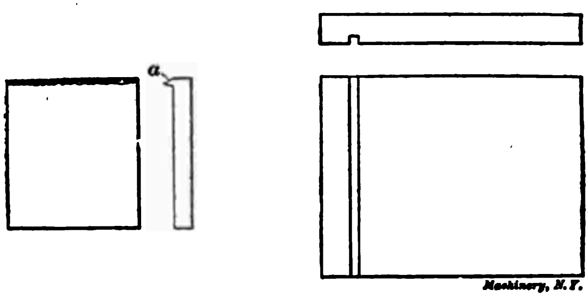
It is necessary when designing tools of any character, whether they be cutting tools or fixtures for holding work while machined, to make provision for the chips. These are liable to get into drill jigs, and despite ordinary care, get under the work or between it and the locating points. In order to do away, so far as possible, with this tendency, it is advisable to cut away as much of the seating surface as can be spared, and to locate stops away from the seating surface, if possible. The seating surface should be smooth enough so that chips will not adhere to it, and so that waste will not stick to it, but it should not be a polished surface, as we would in all probability get it out of true, if we attempted to polish it. If chips are allowed to get under the work it will not be drilled true; that is, the holes will not be at the proper angle with the working surface, and consequently the piece will be unfit for most purposes.

Many operations of machining are almost sure to throw a burr on one side of the piece, and in shops where quantities of work of the same kind are machined, many employes are kept busy removing these burrs in order that they may not interfere with the proper seating of the pieces during the succeeding operations. While the operation of removing the burr on a single piece of work may not incur great cost, yet when thousands of pieces are machined each day, the aggregate cost constitutes quite an item of expense, and the successful manager

is he who so far as possible eliminates the small items of expense, knowing that many small items of expense amount to a large item in the aggregate. Not only is the operation of burring expensive, but as the class of help usually employed to do this work is unskilled, surfaces are many times left in a condition anything but satisfactory. As a consequence, the surfaces of jigs, milling machine fixtures, etc., are many times cut away to receive these burrs, thus doing away with the necessity of burring, as it many times happens that subsequent operations remove the burrs. In Fig. 1 is shown a piece of work having a burr thrown up at *a*, while Fig. 2 represents a surface cut away to receive the burr.

Factors Determining the Advisability of Using Jigs

When we wish to drill two holes a given distance apart, the location of the holes is obtained by means of a pair of dividers set to a scale. The location is obtained and prick punched, after which the holes are drilled. This method answers nicely when one piece is to be drilled, and precise measurements need not be observed. If it is necessary to drill ten thousand pieces, then this becomes a costly method, and the work can be done more cheaply if a jig is made to hold the pieces. The jig must, of course, have holes the size of the drill, which are properly located. By the use of the jig, the cost of drilling is but a fraction of what it would be if the holes were located by dividers, and the surface prick punched as described. As we have already said, the first factor which must be considered is the cost of the jig. If the cost of the jig, plus the cost of drilling, would exceed the cost if the pieces were first prick punched and drilled as formerly described, then the making of the jig would not be considered unless a greater degree of accuracy was necessary than would be liable to be the result of the method mentioned. When a jig is to become a permanent part of the equipment of a shop, its first cost is not so much a matter of consideration as when only a limited number of pieces are to be drilled. Yet no unnecessary expense should ever be allowed.

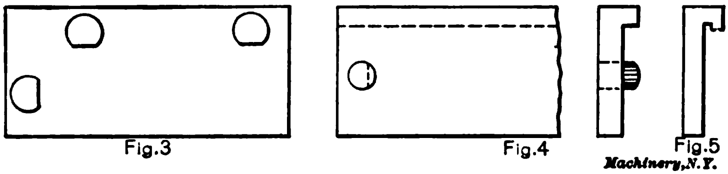


Figs. 1 and 2. Work with Burr, and Grooved Part of Jig to Correspond

Means for Locating Work in Jigs

Many times when only two pieces are to be drilled which must be exactly alike as regards location of holes, it is cheaper to make a

simple jig than to attempt to drill them by any of the methods commonly used in machine shops. In such a case the jig may be made from a piece of cast iron or other material which may happen to be on hand, the holes being carefully laid off and drilled. This jig makes it possible to drill the holes in both pieces exactly alike as to location. When using a jig of this description it is possible to locate the holes near enough for most work by ordinary measurement. If many pieces



Figs. 3, 4 and 5. Means for Locating Work in Jigs

were to be drilled, it would be necessary to provide locating points, so that the pieces could be placed in the jig, and the essential surfaces brought against these. The means of locating may be pins, as shown in Fig. 3, or a shoulder and a pin, as in Fig. 4. If pins are used, they should be so located that the bearing surfaces may be worked flat, as shown, to prevent wear, and also to do away with a tendency to press into the surfaces of the work. If flat shoulders are used they should be cut away, or relieved, at corners, as shown in Fig. 5, to do away, so far as possible, with the liability of dirt or chips getting between them and the work. Then, again, if the working edges of the pieces of work are not exactly true, it would be impossible to properly locate by pressing them against true locating surfaces which extend the whole length.

When work is of irregular contour that could not be properly located by bringing it against two locating surfaces, it is possible to provide a locating device which bears against all the surfaces, as shown in Fig.

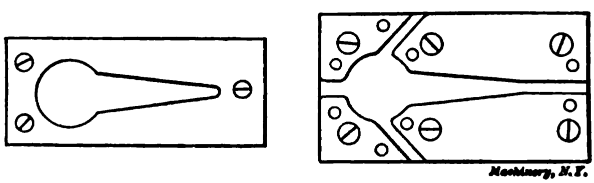


Fig. 6. Method of Locating Work in Jigs

6. This method, however, is hardly to be advocated for most work, as it necessitates exactness of measurement and shape on all the bearing surfaces, as well as on the pieces to be worked upon. Then, again, the shape makes it extremely difficult to clean, and a chip under any portion of the work will cause it to stand at an angle with the seating surface of the jig.

Clamping Devices

It is necessary to hold the work solidly in the jig without any chance of its changing its location. Should the location change after one or

more holes are drilled, and before all are drilled, it would cause a variation that would in all probability spoil the piece of work. When but a few pieces are to be drilled with a jig it is not generally considered advisable to make jigs with fastening devices, the work being held in place with a clamp, as shown in Fig. 7. In order to do away with any possibility of change of location, a pin is forced through the jig hole and the hole in the work after drilling the first hole. If many holes are to be drilled in a piece it is advisable to have two pins. After drilling a hole in one end of the piece, force in a pin; then drill a hole in the opposite end, and place a pin in this hole, as shown in Fig. 8. The pins in opposite ends of the piece will prevent its slipping when the rest of the holes are drilled. Many different forms of fastening devices are provided, the design depending on the class of work. One of the most positive methods consists of a screw which passes through a stud or some elevation on the jig, and presses

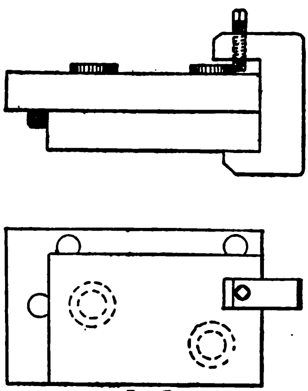


Fig. 7

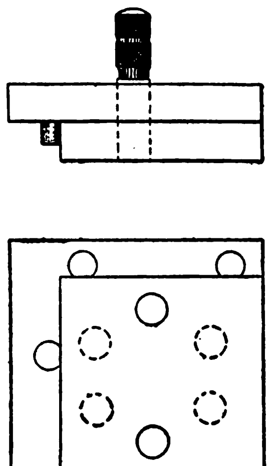


Fig. 8 *Machinery, N.Y.*

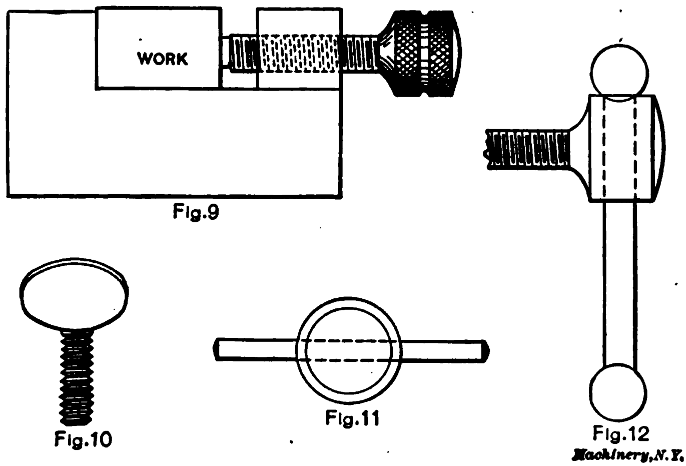
Figs. 7 and 8. Means for Holding Work in Drill Jigs

against the work, forcing it against the locating points, or stops, as they are called. The screw may have a knurled head, as shown in Fig. 9, or a thumb-screw may be used, Fig. 10. Sometimes it is necessary to exert greater pressure than can be applied by means of a screw of the ordinary form. Then, it is possible to make a screw with a round head, drill a hole through it, and through this hole pass a piece of wire as shown in Fig. 11. By this screw, sufficient pressure can be applied. When it is necessary to exert a greater amount of power than would be possible by the use of a pin of the length shown in Fig. 11, one may be used that will slide freely in a hole in the head of the screw. A ball placed on each end prevents its falling out. By getting the full length of the pin on one side of the screw-head, as shown in Fig. 12, a much greater amount of power is obtained. At times the stud which supports the screw may interfere with the plac-

ing of the work in, or the removal of the work from the jig, or it might be necessary to turn the screw for a considerable distance each time the work was placed in or taken out of the jig. In such cases a stud could be provided that could be removed from the jig when the screw was relieved of its tension against the piece of work. Such a stud is shown in Fig. 13.

Clamping Work by Cams or Eccentrics

A common method of fastening work is by means of a cam of suitable form. Cams of the ordinary design are not as powerful as the screw, but they have the advantage of being more quickly operated,



Figs. 9 to 12. Means for Clamping Work in Drill Jigs

and in the case of light work where but little strength is required, they answer the purpose much better. The designer should bear in mind that a few seconds' time saved on each piece of work amounts to a large saving in a day when a number of hundred pieces are placed in and taken out of a jig; and in these days of competition every means of saving time consistent with quality of work should be considered. When the work bears against two points—one on the side and one on the end—the cam should be designed so that its travel against the work will force it against both, rather than away from one. Fig. 14 shows a piece of work held by a cam which, by means of the handle, forces the work inward and in the direction of the arrow, thus holding it against the locating pins *a a* and the end stop *b*. In order to get as much pressure as possible with a cam, it is necessary to have the portion that bears against the work when it is against the locating surfaces nearly concentric with the screw hole. This being the case, it is obvious that the pieces must be very nearly of one size, while in the case of a screw binder any amount of variation may be taken care of. Thus it will be seen that a screw may be used where a cam would not answer. However, it is advisable to use a cam in prefer-

ence to a screw when possible, but at times the piece of work may be subjected to repeated jars which would tend to turn a cam, thus loosening the work. In such cases a screw is preferable. If a cam would be in the way when putting in or taking out work, it may be made removable, as shown in Fig. 15. At times a tapered piece of steel in the form of a wedge may be used to hold work, as shown in Fig. 16.

Simple Forms of Drill Jigs

When many pieces are to be drilled in a jig made in the simple form shown in Fig. 17, the drill wears the walls of the holes, enlarging

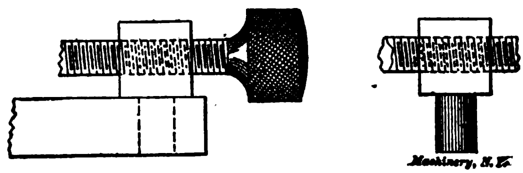
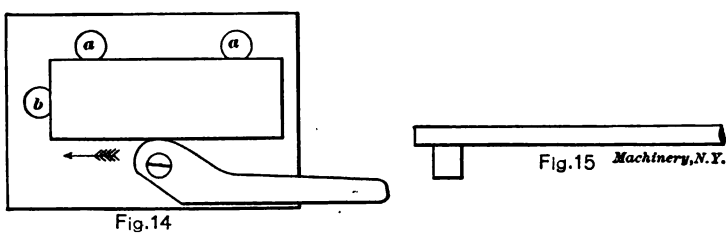


Fig. 13. Clamp Screw Mounted in Removable Stud

them sufficiently to render accuracy out of the question. Where jigs are to be used enough to cause this condition, the stock around the walls of the hole may be hardened, if the jig is made from a steel that will harden. If made from machinery steel, the stock may be case-hardened sufficiently to drill a large number of pieces without the



Figs. 14 and 15. Eccentric Clamp for Simple Drill Jigs

walls wearing appreciably. This, however, would not answer when accuracy is essential, as the process of hardening would have a tendency to change the location of the holes.

Guide Bushings

When the jig is to be used for permanent equipment, or where many holes are to be drilled, it is customary to provide bushings—guides—made of tool steel and hardened. These are ground to size after hardening, and being concentric, may be replaced, when worn, by new ones of the proper size. It is the common practice to make bushings for drill jigs on the same general lines as shown in Fig. 18, the upper end being rounded to allow the drill to enter the hole readily. A head is provided, resting on the surface of the jig; the portion that enters the hole in the jig is straight, and is ground to a size that insures its remaining securely in place when in use.

If the hole is sufficiently large to admit a grinding wheel, it is

ground to size after hardening. In such cases it is, of course, necessary to leave the hole a trifle small—0.004 inch—until it is ground. If the hole is not large enough to allow of grinding, or if there is no means at hand for internal grinding, the hole may be lapped to size by means of a copper lap, using emery or other abrasive material, mixed with oil. When the hole is to be lapped rather than ground,

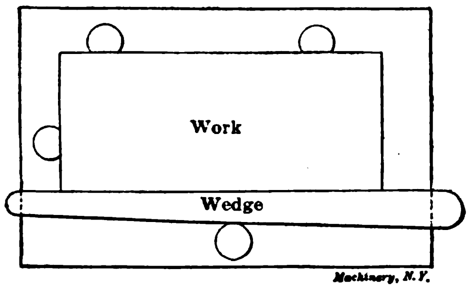


Fig. 16. Wedge Acting as Clamp in Drill Jig

leave a smaller amount of stock to be removed by the operation, say 0.001 inch or 0.0015 inch. After grinding or lapping the hole to size, place the bushing on a mandrel and grind the outside until it is a pressing fit in the hole. While on the mandrel, be sure to grind the

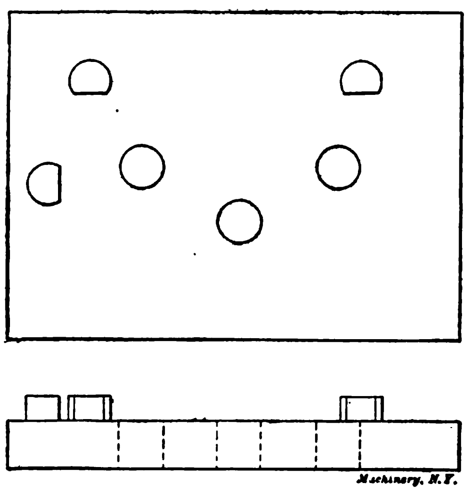
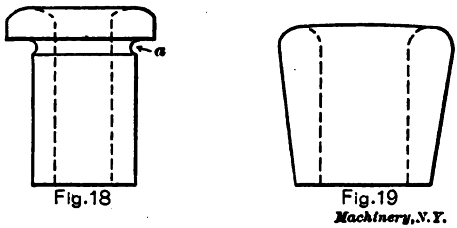


Fig. 17. Simple Form of Drill Jig without Bushings

under portion of the head, *a*, Fig. 18, to insure its being true with the body. Before starting to grind the outside of the bushing, test the mandrel for truth. This should be done *after* placing the bushing on it rather than before.

It is the custom in a few shops to make the outer portion of bushings tapered, as shown in Fig. 19. Unless there is a sufficient reason for so doing, this is to be avoided, as the operation of making a tapered

hole, unless it is bored on the taper with an inside turning tool, is not likely to produce a hole, the axis of which is at the desired angle to the surface of the jig. The outer portion of the bushing can easily be ground to the desired taper, but there is the liability of a particle



Figs. 18 and 19. Bushings for Drill Jigs

of dust getting in the hole when placing the bushing in the jig. A tapered bushing, in order to get the proper taper, necessarily costs a great deal more than a straight one, and cannot answer the purpose any better, and probably not as well.

Types of Drill Jigs

The shape and style of the jig must depend on the character of the work, the number of pieces to be drilled, and the degree of accuracy essential. It may be that a simple slab jig of the design shown in Fig. 20 will answer the purpose; if so, it would be folly to make a more expensive tool. If we are to drill a piece of work of the design shown to the left in Fig. 21, and but one hole is to be drilled in each piece, then a jig made in the form of an angle iron, as shown to the

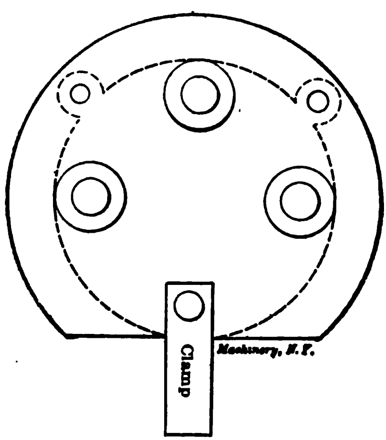


Fig. 20. Slab Jig of Simplest Design

right in Fig. 21, works nicely, and is cheaply made. As it is not necessary to move the jig around on the drill press table it may, after locating exactly, be securely fastened to the table. In designing such a jig, it is advisable, when possible, to have the work on the side

of the upright shown in Fig. 21, rather than on the opposite side, as that does away with any tendency of the jig to tip when pressure is applied in the operation of drilling.

Leaf Drill Jigs

For many kinds of work a jig provided with a leaf, as shown in Fig. 22, gives best results, as the leaf may be raised, and the work removed, and any dirt cleaned from the working surfaces. After placing the piece to be drilled in the jig, the leaf is closed. As the bushings are in the leaf, it is apparent that it must always occupy the same relative position to the work for the different pieces, or they will not be duplicates; consequently, the fulcrum pin, *a*, must be a perfect fit in the hole in the leaf, and a locating pin *b* is provided to prevent any tendency of the leaf to move from the action of the drill when cutting. Jigs provided with such a pin show less tendency to wear in the joint. The leaf should not close down onto the work, but

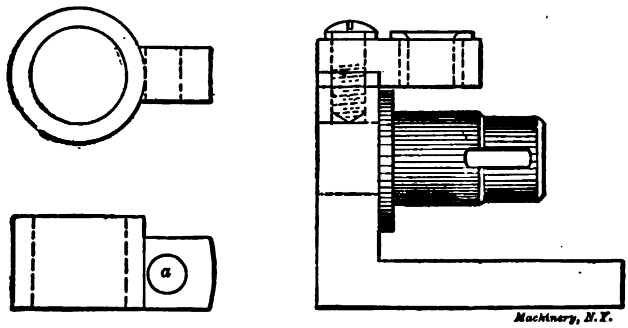


Fig. 21. Piece to be Drilled, and Jig Used for this Work

onto a shoulder on pin *b*, as shown, there being a space between the work and the jig leaf.

While the above is true for most work, a jig for drilling round pieces may be designed as shown in Fig. 23, the holding device being two V-shaped blocks, one located on the lower portion of the jig, while the other is on the leaf, as shown. In the case of a jig of this pattern, the work is securely held by binding the cylindrical piece by pressing the handles of the jig together.

Jigs Provided with Feet or Legs

When jigs are to be moved around on the table of the drill press, as is the case where several holes are to be drilled, feet or legs are generally provided, as shown in Fig. 22. In order that the legs may not wear, it is customary to harden them. The legs are hardened before they are placed in the jig, and are ground and lapped true while in the jig. As the only wear is on the ends, or where they come in contact with the drill press table, it is customary to harden only the ends which rest on the table. In most shops jig legs are made from tool steel, although a good grade of open-hearth steel containing sufficient carbon to insure its hardening answers as well for most purposes. But

as few shops carry such steel in stock, crucible tool steel is generally used. The ends of the legs should be ground true with the seating surface—that is, where the work rests—of the jig. To accomplish this a surface grinder should be used. As the operation of grinding leaves a number of projections on the surface ground, and as these ridges or projections would wear away as the legs were moved back and forth

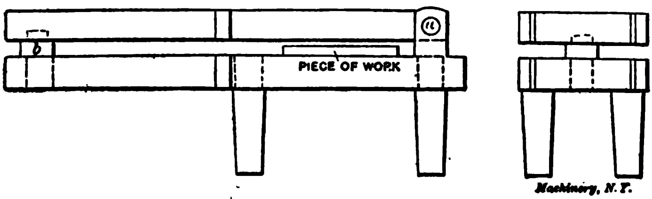


Fig. 22. Jig with Pivoted Leaf

on the drill press table, it is advisable to remove them by lapping on a flat lap, thus producing a perfectly smooth, true surface. In this way we reduce the wear to a minimum.

For certain classes of jigs the legs may be short, not more than ½ inch long; but for jigs of the style shown in Fig. 22, where the tool is held in the hand, it is necessary to make the legs longer to

Fig. 23. Part of Jig with Pivoted Leaf, Showing Method of Holding Round Work

keep the fingers from coming in contact with the chips on the drill press table. The legs should be located so as to do away with any tendency of the jig to tip up when the work is being drilled.

Relation Between Accuracy of Jigs and Accuracy of Machines on which They are Used

While it is necessary to observe extreme care in designing drill jigs to prevent any tendency of the jig to tip, and to have the legs ground and lapped on a true plane, it is just as necessary that the drill press table should be perfectly at right angles to the spindle, and that it should be true and flat. Otherwise, the holes will not be at the desired angle with the working surface of the work.

In shops where interchangeable work is produced, or where the work must in all respects be machined correctly, the condition of the various machines is closely watched, and especially such parts of the machines as affect the accuracy of the finished product. Drill press tables are planed over when out of true, or are lined up to insure their being at right angles to the spindles of the drill press. This may be done by placing a bent wire in the drill chuck, the wire being bent so that it will describe as large a circle as possible, and yet be free to swing. The end of the wire is bent so that the point will come in

contact with the table. By loosening the screws holding the table, and inserting "shims," it may be trued as desired.

Locating the Holes for the Drill Bushings

When making jigs, the part of the work that calls for the best workmanship is locating the holes for the drill bushings. The methods employed differ, but should depend on the character of the work. Where accuracy is not essential, it is the custom many times to take a piece of work that is right, or, rather, one where the holes are drilled near enough right, place this in the jig and transfer the holes into the jig. As it is necessary to leave the bushing holes in the jig considerably larger than the holes in the work in order to have sufficient stock

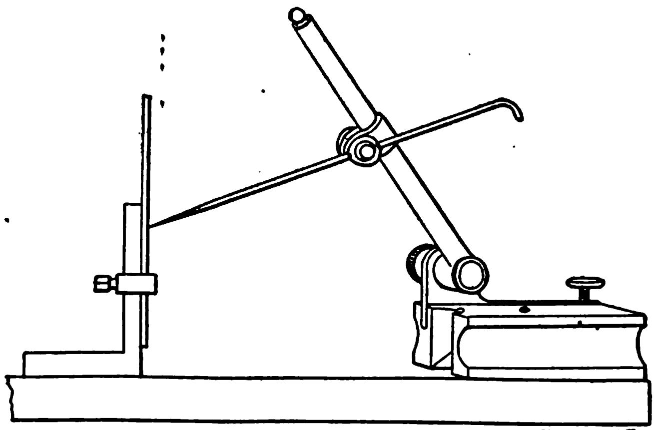


Fig. 24. Method of Taking Vertical Measurements

around the holes in the bushing, those in the jig may be enlarged by means of a counterbore, the pilot of which fits nicely in the transferred holes, and with a body the size of the desired hole. When this method does not insure desired accuracy, several other methods may be employed.

Making a Jig from a Sample Piece or Model

If a model of the work to be done is at hand, a jig, as shown in Fig. 22, may be made in the following way: The leaf is raised and the model put in place. The jig is fastened to the face-plate of the lathe, the leaf still being raised. By means of a center indicator the jig is located so that one hole of the model runs true; the leaf is then closed and the hole is drilled through it, and then bored with a boring tool to the desired size. Never ream a bushing hole in a jig, or any similar hole in any piece of work, where the finished hole must be exactly located, as a reamer is liable to run out somewhat and thus affect the accuracy of the work. A reamer, if properly made and used, will produce a round, true hole, accurate as to size, and is a valuable tool for many purposes, and holes of a uniform size may be produced. But on account of the stock being uneven in texture, or on account of

blow holes in castings, a reamer is liable to alter its course and so change the location of the hole. While for many purposes this slight alteration of location might be of no account, yet for work where accuracy is essential, it is out of the question.

After drilling and boring the first hole, the jig may be moved on the face-plate, and the other holes produced. It is abvious that in order to produce holes that will be at right angles to the base of the jig, the face-plate of the lathe must run true, and should be tested each time it is used for any work where accuracy must be observed.

Method of Locating Holes When Accuracy is not Essential

Where there is no model, and it is not considered advisable to make working models of the various parts, the location of the bushing holes may be obtained by laying out the various points on the jigs. In such cases a drawing is usually furnished, and the dimensions on same are transferred to the face of the jig. If it is not necessary to have the holes exact as to measurements, the laying out may be done with a surface gage, the point of the needle being set to a scale. The scale

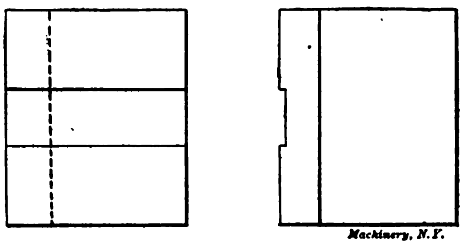


Fig. 25. Angle Iron with Groove for Scale

may be clamped against an angle iron, as shown in Fig. 24, or an angle iron may have a groove of the width of the scale cut across its face at right angles to the base, as shown in Fig. 25. The scale should be a good fit in the groove, so fitted that it will stay securely at any point from frictional contact with the sides of the slot, or a spring may be so arranged as to insure the proper tension.

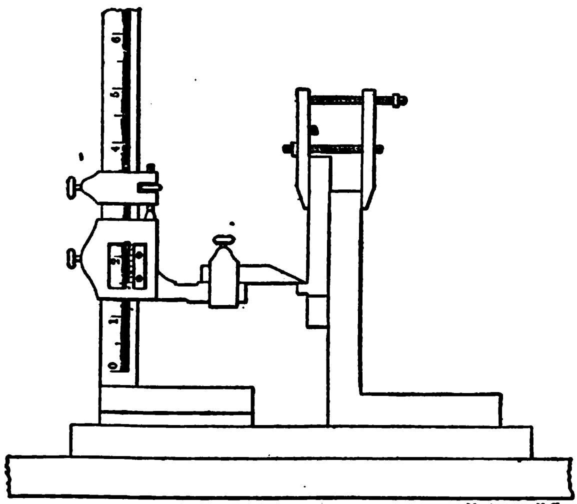
Method Assuring a Fair Degree of Accuracy

Where greater accuracy is essential, the working points should be obtained by means of a height gage, as shown in Fig. 26. By means of such a tool the measurements may be fairly accurate, as the vernier scale allows of readings to one-thousandth inch. When the lines have been scribed at the proper locations they are prick punched. In order to prick punch exactly at the intersection of lines the operator must wear a powerful eye-glass, and use a carefully pointed punch, ground to an angle of 60 degrees. If the punch marks are made very light at first, the exact location may be observed nicely. The punch marks should not be deep, as there is a liability of alteration of location if the punch is struck with heavy blows. After the various points have been located and punched, the jig may be clamped to the face-plate of the lathe, and the bushing holes carefully drilled and bored to size.

At times jigs are made of such size and design that it seems wise to core the bushing holes. In such cases it is necessary, in order that we may lay out the location of the centers of desired holes, to press a piece of sheet steel or sheet brass into the cored hole, as shown in Fig. 27, and locate the center on this piece. When the holes are properly located for machining, the sheet metal may be removed and the holes finished to the desired size. If an error of 0.001 or 0.002 inch is not permissible, the method described above should not be employed.

Method Employed for Highest Degree of Accuracy

Where extreme accuracy is essential we must locate round pieces of steel on the face of our work. These pieces of steel are called buttons and are of exact size and perfectly round. To do away with any possibility of their becoming bruised in any way, they are hardened and carefully ground to size. The buttons are attached to the work by means of machine screws, as shown in Fig. 28, the holes in the but-



Machinery, N.Y.

Fig. 26. Taking Vertical Measurements by Means of Height Gage

tons being larger than the screws used; this difference in size allows us to move the button until it is accurately located. The diameter of the buttons should be some standard size, exactly divisible by two, because, in making our computations we only consider the distance from the center of the button to its circumference, that is, the radius.

When we start to lay out the centers for the bushing holes we first determine our working surface, then lay out on the face of the jig, by means of a surface gage, as described in a previous operation, the centers of the holes to be produced. We then drill and tap screw holes to receive the screws to be used in holding the buttons to the jig. When we have prick punched the surface, and before drilling the holes, we scribe by means of dividers a circle the size of the button on

the face of the jig with the punch mark as center. This enables us to approximately locate the button. If the hole to be produced has its center 2 inches from the base *a* and 4 inches from vertical side *b*, Fig. 29, we would locate the button—provided it was ½ inch diameter—1¾ inches from *a*, and 3¾ inches from *b*. This can be done accurately by the use of a vernier caliper, or we can lay the jig on the side *b*, and by means of a length gage, or a piece of wire filed to the right length, accurately determine the distance from *b* to the button. The jig is then placed on the base *a* and the other dimension obtained in the same manner. The buttons may be located more easily by the use of a vernier height gage, if one is at hand.

If there are to be several bushings on the face of a jig, a button may be accurately located where each hole is to be. The jig may be clamped to the face-plate of the lathe so that one button is located to run exactly true. This is done by means of a lathe indicator. When

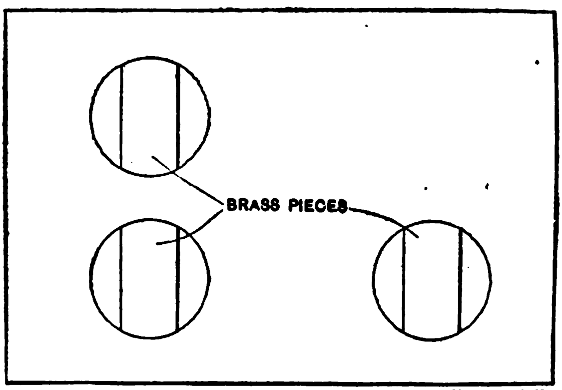


Fig. 27. Cored Holes with Inserted Brass Pieces for Centers

the jig has been so located that the button runs perfectly true, the button may be removed and the hole enlarged by means of a drill, so that a boring tool can be used to bore it to the proper diameter.

Locating the Holes on the Milling Machine

In some shops it is not considered advisable to locate a button at the desired position of each bushing hole. One button is located and the jig is fastened to the table of a milling machine having a corrected screw for each adjustment. Then, after one hole is accurately located and bored, it is a comparatively easy matter, by means of the graduated dials, to obtain the other locations; however, this method should never be used unless the machine has all its movements governed by "corrected" screws, as the screws ordinarily sent out on milling machines are not correct as to pitch, and if used, serious defects in measurements will result. Many tool-makers, therefore, prefer using a vernier scale and vernier attached to the knee and table of the milling machine, for accurate work, as they are then independent of the inaccuracies that may be present in the feed-screw.

Fig. 30 shows a jig clamped to an angle iron on the table of the milling machine. The angle iron is located exactly in line with the travel of the table, and the jig fastened to it. The button *D*, which has previously been accurately located, serves as a starting point, and the jig must be located so that the button is exactly in line with the spindle of the machine. This is accomplished by moving the table

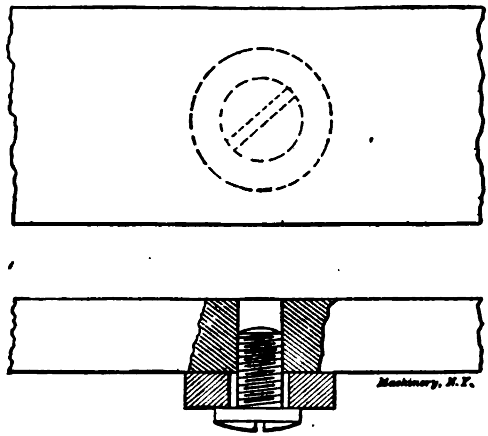


Fig. 28. Buttons for Locating Holes in Jigs

until the sleeve *A* on the arbor *B* will just slide over the button *D*. The hole in *A* must be a nice sliding fit on the arbor *B* and also on the button *D*. In order to insure accuracy, the arbor *B* must be turned to size in the spindle just as it is to be used; or, if a portable grinder is at hand, the arbor may be fitted to the spindle hole or to the collet, as the case may be; the portion which receives the sleeve *A* may be

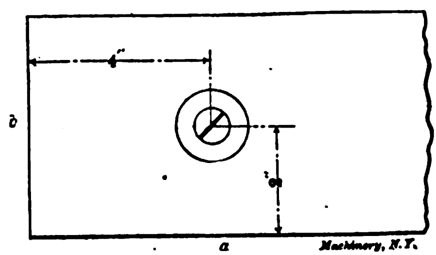


Fig. 29. Locating a Hole by Means of a Button

left a trifle large, and may be ground to size in place on the machine. The portable grinder is located on the table of the machine.

After the jig has been accurately located so that the button *D* allows the sleeve *A* to slide over it, the arbor *B* may be removed from the spindle, and a drill be employed to increase the size of the tapped screw hole that received the screw used in fastening the button. Best results follow if a straight-fluted drill, as shown in Fig. 31, is used. The drill should not project from the chuck or collet any further than necessary,

thus insuring the greatest rigidity possible. After drilling, a boring tool of the form shown in Fig. 32 may be substituted for the drill, and the hole bored to size. The machine may now be moved to position for the next bushing hole by observing the dimensions given. The operator should bear in mind that the screw used in getting the spac-

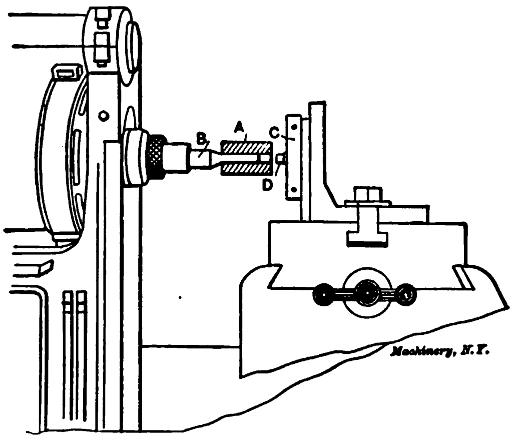
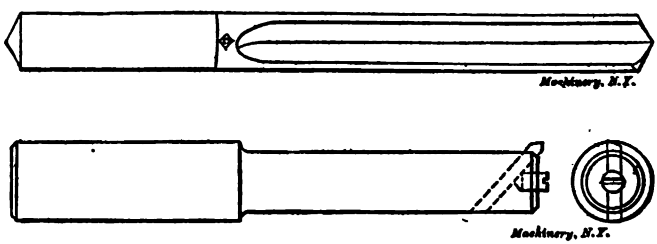


Fig. 30. Locating Holes in the Milling Machine

ings must be turned in the same direction at all times, otherwise the backlash will render accuracy out of the question.

While the foregoing relates to plain jigs, the same principles apply to those of more complicated design. In the next chapter attention



Figs. 31 and 32. Straight-Fluted Drill and Inserted Cutter Boring Tool

is given to a different and original method of locating the holes in jigs, using the drill press for this work exclusively, and Chapter III is devoted to examples of actual designs of drill jigs, showing how the elementary principles outlined above are employed in the practice of the machine shop.

CHAPTER II

DRILLING JIG PLATES*

A description of the following method of drilling jig plates was contributed to the columns of MACHINERY, October, 1902, by Mr. J. R. Gordon. The method being radically different from any of those in common use, it has been deemed proper to mention this method in connection with other methods for locating the holes in jigs already referred to.

In the case in question, a great many jigs were to be made, and the positions of the drill bushings were to be accurate within 0.001 inch. The procedure was as follows: The regular work-table from an ordinary sensitive drill press of the usual pattern was removed, and substituted by one of larger dimensions, as this was called for by the size of the jig plates to be made.

This table was first planed on the face and edges, and the stem, by which it is held in the bracket on the column of the press, was turned to fit snugly the hole in the bracket. After planing and turning the table, a series of holes was drilled, as shown in Fig. 34, and they were tapped to receive a No. 14-20 screw. Two parallel pieces *C* and *D*, Fig. 34, having straight edges and a thickness of 3/8 of an inch, were made. These may be clamped to the table in such a position as may be desired or the work determine, the series of holes permitting any adjustment within the range of the table. In order to make more room between the spindle and the column of the drill press, the spindle head was blocked out, the block having a projecting lug, as shown at *A*, Fig. 33, to which a bracket, *F*, was fastened to carry the bushing, *B*. This bushing is fastened by a screw and can readily be removed and others inserted, having various sizes of holes, if found desirable. These preparations were all that were necessary with the exception of the gages that will be described in the operation of the method for spacing, which is as follows:

The plate to be drilled had a number of holes spaced as shown in Fig. 34, and before drilling them they were marked as Nos. 1, 2, 3, etc., No. 1, as will be seen, being the upper left-hand hole. Its location with reference to either end or sides of the plates did not require to be very exact; but other plates may need to have holes placed at some definite distance from the edges or ends, so it may be assumed that the distance is 6 inches from the edge, *G*, and 8 inches from the end, *H*.

With these distances given, make two gages, using vernier or micrometer calipers for standard, and make them 6 1/8 and 8 1/4 inches long, respectively. Remove the bushing, *B*, Fig. 33, and in its place insert a plug having a diameter of 1/4 inch.

* MACHINERY, October, 1902.

Resting the 6⅛ gage on the table, and with one end touching the plug, the parallel piece, *C*, Fig. 34, is brought to just touch the other end of the gage and is then clamped to the table. This is not very difficult if one end of the parallel is left free and the other end is clamped tight enough to permit the free end to move somewhat stiffly. After locating and clamping the parallel, *C*, the other parallel

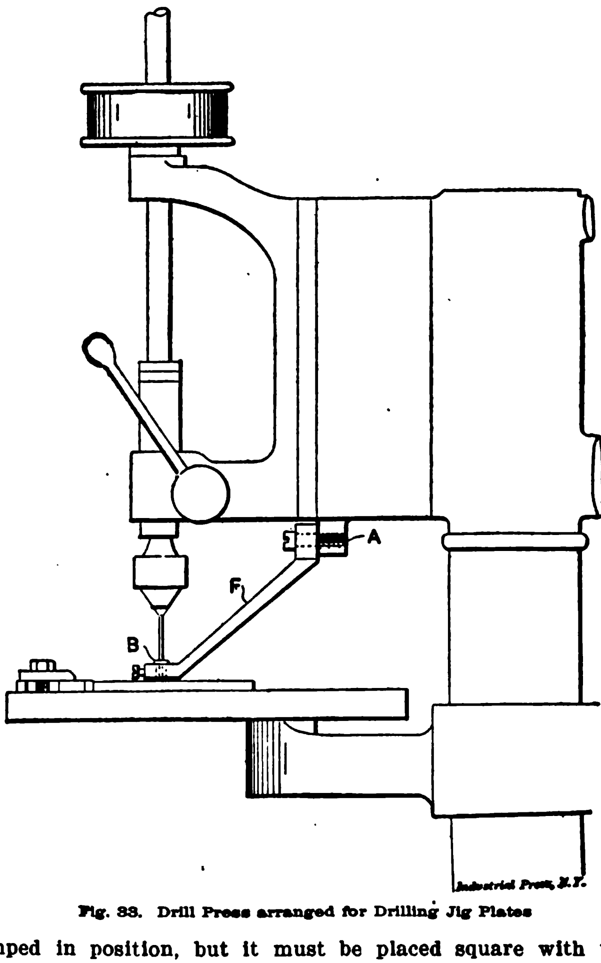


Fig. 33. Drill Press arranged for Drilling Jig Plates

is clamped in position, but it must be placed square with the first parallel. This is more difficult than in the first case, but is not at all difficult if one man can be employed to clamp the piece while another holds the square and gage. The reason for making the gages 6⅛ and 8⅛ inches long instead of 5⅞ and 7⅞ inches, respectively, is that it is not desirable to have the edges of the plate touch against the parallels, as chips could get between the two and destroy the accuracy of the measurements; allow the gage to be ¼ inch longer than the distance

required, and fill in the space with ¼ inch diameter gages, as shown in Fig. 34.

For gages over 1 inch in length use flat brass rods or strips about ⅝ inch wide and ⅛ inch thick, and cut them a little longer than the

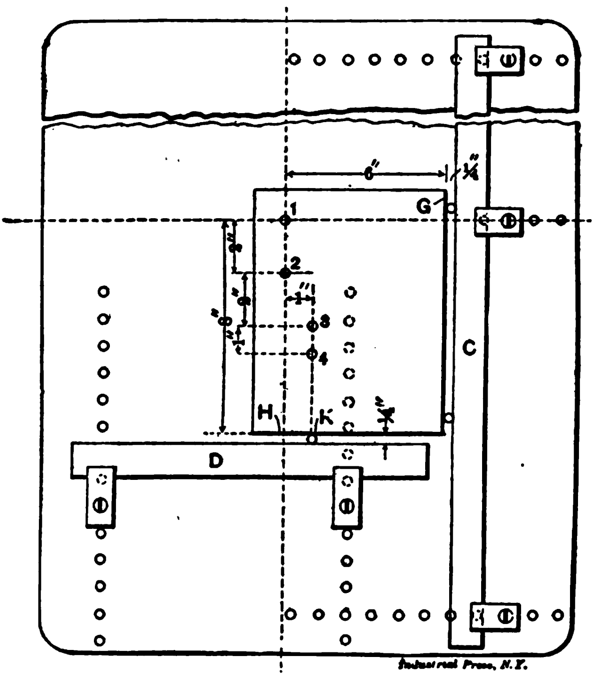


Fig. 34. Plate in Position for Drilling First Hole

finished length. One end is finished square and the other end is rounded as shown in Fig. 35. In making the gage, if too much metal is removed, it is an easy matter to pene the stock out to make up for any reasonable error. The length of the gage is stamped on it, and when the operation is completed it is put away for future use.

Having located the parallels, the plug is removed from the bracket

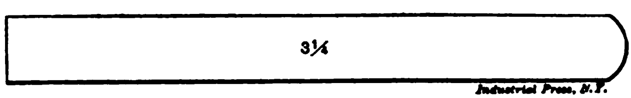


Fig. 35. Type of Gage used when Drilling Jig Plates

and the bushing replaced. The drill should, of course, fit as snugly to the hole in the bushing as it can and run without cutting. The bushing should support the drill to within a distance equal to the diameter of the drill from the plate to be drilled, and care should be taken not to drill through the plate until all the holes have been started. After drilling the first hole, to place the plate for the second hole, distant 2 inches from the first, it is moved along the parallel, *C*,

and a gage 2¼ inches long placed as shown in Fig. 36, and when so placed is ready for drilling. The third hole requires three new gages, since it is 1 inch off the line of the other two holes, as shown in Fig. 37.

For holes which are to be finished 3/16 inch to ¾ inch in diameter, use first a small drill, size No. 52 to No. 30. After the holes are all drilled to this size, then enlarge them, by the use of a series of four lip counterbores, to the required size. Where extreme accuracy is required, in the place of the counterbore, a small boring bar may be substituted and the holes bored to the size desired. One disadvantage of using a boring tool is that it requires a hole in the table equal to the largest hole to be bored out, or that the plate shall be kept clear of the table by blocking up with parallel strips under it.

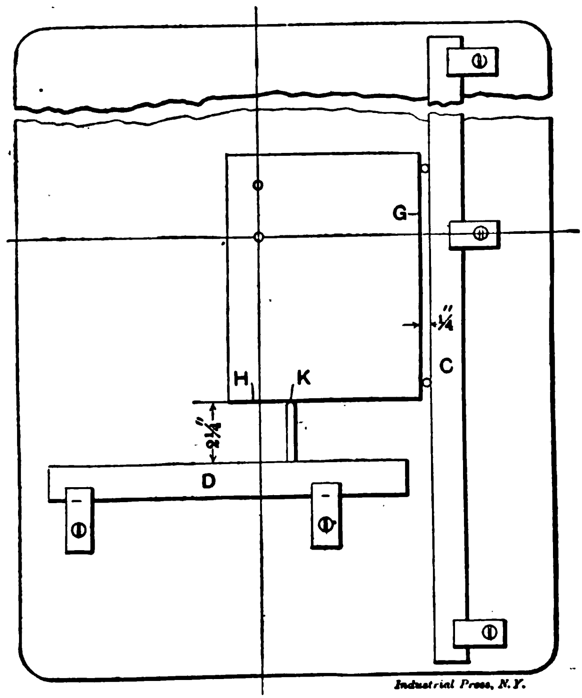


Fig. 36. Plate in Position for Drilling Second Hole

Fig. 38 shows a form of boring tool which will be found very convenient for use on this kind of work. It consists of the shank, *A*, which is fitted to the taper hole in the spindle, and a split holder, *B*, which is pivoted to the shank at *C*, and is locked to it at *D*, the screw at *D* serving to clamp the boring tool, *E*, at one end, while *F* clamps it at the other end. Adjustment is obtained by swinging the holder, the radial slot, *G*, allowing it to have quite a range, and the top screw, *H*, permitting fine adjustment. Split bushings in the holder will allow the use of boring tools of smaller diameter if desired.

This method of locating holes is not limited to the drill press, but may be employed to advantage on the face-plate of a lathe. In this case, the work, as soon as located by the gages, is clamped to the face-plate.

While this method was originated for drilling holes in jig plates, it may be used with equal success for drilling small interchangeable pieces. It is not necessary that the edges, *G* and *H*, be planed at right angles, as the same results will be obtained if the surface, *G*, is planed true and a finished spot provided at *K*, from which point all measurements to the parallel, *D*, are made.

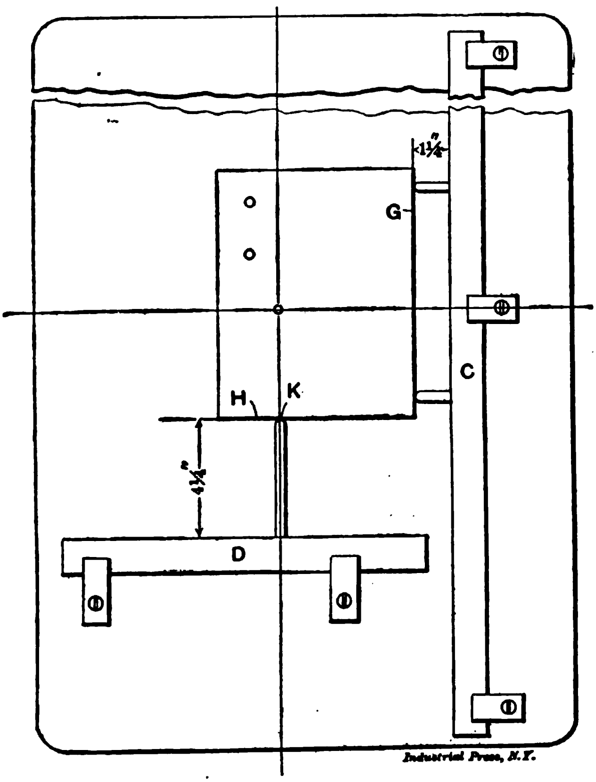


Fig. 37. Plate in Position for Drilling Third Hole

Mr. Gordon claims that this system has certain advantages over the button methods used on the milling machine. In the first place, the feed-screws on nearly all milling machines are not correct, and in some shops the tool equipment is so badly worn as to make the use of the feed-screws out of the question for accurate work.* However accurate a screw on a milling machine is when new, it soon loses its truth under ordinary conditions of machine shop practice, since only a small

* See page 18: Locating the Holes on the Milling Machine.

portion of the screw is used to do most of the work of driving the table. In the second place, Mr. Gordon claims that his method is quicker, the supposition being that the necessary appliances, such as parallels, brackets, bushings, etc., are made and ready for use; and finally, that there is a very small chance for errors, provided that the gages used are marked distinctly.

These assertions, however, called forth considerable comment in the columns of MACHINERY. Mr. Frank E. Shailor, in particular, took issue with Mr. Gordon on account of these assertions and claimed that there were considerable chances for errors. Mr Gordon, however, defended

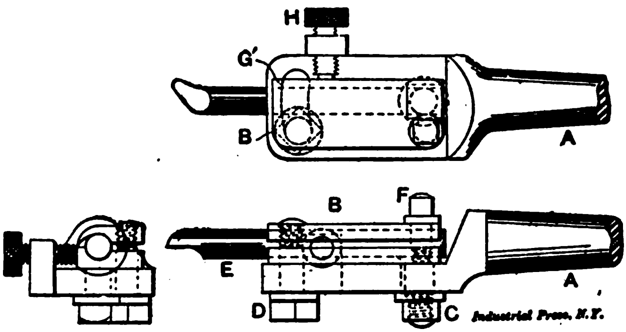


Fig. 38. Boring Tool

his method, pointing out that most of Mr. Shailor's objections were of little consequence, provided proper precautions were taken. Other contributors added their word to the discussion, some siding with Mr. Gordon, and some admitting that the methods used both by Mr. Gordon and Mr. Shailor would, under proper circumstances, be correct to use. It is not possible in this treatise to give place to what was more a personal controversy, than of direct bearing upon the subject of drill jig design. It may, however, be proper to mention that the discussions on this subject appeared in the July, August, September and November, 1904, and the January and February, 1905. issues of MACHINERY.

CHAPTER III

EXAMPLES OF DRILL JIGS

In the following will be given a number of examples of drill jig designs for definite purposes, as employed in various shops in the country. No attempt has been made to show only jigs of which it can be said that the design is perfect or nearly so, but examples have been taken which indicate general practice, and attention has been called to wherein these jigs conform to the principles of drill jigs as treated in Chapter 1, and also to the objections that might be raised against each particular design, if such objections have been considered in

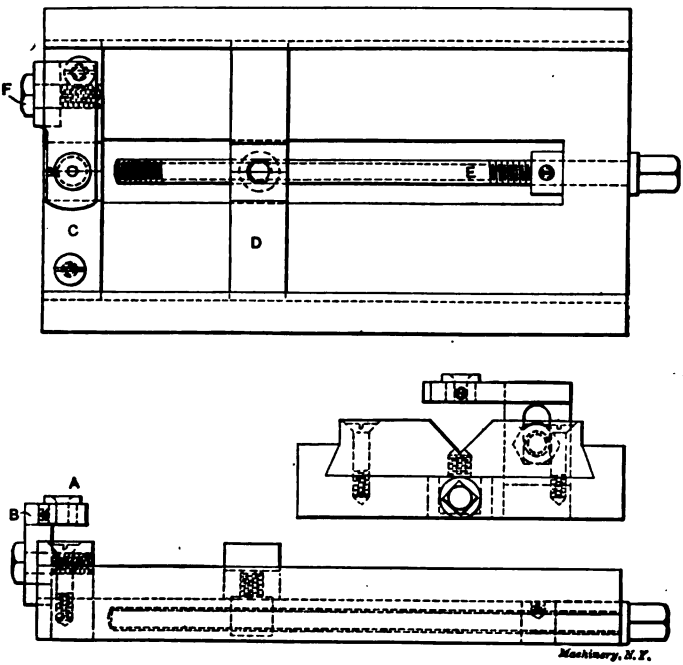


Fig. 39. Jig for Drilling Holes in Studs and Shafts

place. The names of the persons who originally contributed to the columns of MACHINERY the descriptions of the devices shown, have been given in notes at the foot of the pages, together with the month and year when their contribution appeared.

Jigs for Drilling Pin Holes in Shafts

Usually, the simplest kinds of jigs are those intended for drilling a hole through the center of a shaft. They often consist only of a

V-block, in which the work rests, and a cover of the simplest design, containing the guide bushing. Sometimes, however, they are made more universal; the cuts Figs. 39 and 40 show two such designs.

The jig in Fig. 39 is intended for drilling pin holes in comparatively short studs, and will handle a variety of such work with great rapidity. The drill bushing *A* can be removed and bushings with different size holes inserted. The bushing holder *B* can be raised or lowered to suit different diameters of work. The V-block *C* is fixed, while block *D* is adjustable by means of the screw *E* for different lengths of studs. By fastening a strap to the device by screw *F*, and providing this strap with an adjustable screw in line with the V's, studs can be gaged from the end instead of from the shoulder, which, when used for gaging, rests against the sides of either of the V-blocks. The manner in which this jig is used lends itself well to a variety of work of all descriptions.*

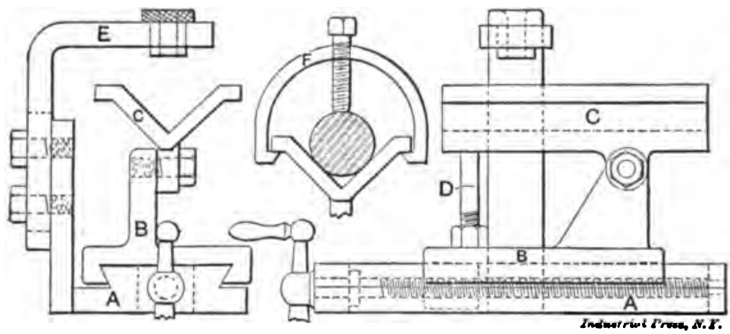


Fig. 40. Jig for Drilling Holes in Shafts

This jig is a simple, yet efficient and characteristic, example of the adjustable type of jig. It will be noticed that the design does not provide for any clamping device for the work to be drilled; this is on account of that in this case the holes to be drilled are so small, compared with the diameter of the shaft or stud, that the stud will stay in place by its own weight, or by pressure of the hand on its upper side, the V-groove aiding materially in keeping the work in position.

The device shown in Fig. 40 is another example of an adjustable jig for this class of drilling. This tool has proved to be of the greatest convenience for drilling shafts, spindles or other round pieces. The base *A* is dovetailed and fitted with a lead-screw, which moves the slide *B* in and out. Upon this slide is mounted the adjustable V-block *C*, which can be tipped at any desired angle for oblique drilling, or set perpendicularly to hold the shafts in position for end drilling. The adjustable stud *D* is placed under the outer end of the block to hold it firmly in any set position. The arm *E* is adjustable up and down, for different sized shafts, and is supplied with a complete set of bushings for use with drills of different diameters. When mortising bars, intended to be used as holders for facers, boring cutters, counterbores

* Paul W. Abbott, August, 1907.

with interchangeable blades, etc., the work is clamped into the V with the clamp *F*, and then, after the first hole has been drilled, the slide is moved along for a distance equal to the diameter of the drill, and the next hole drilled, and so on. By this method any number of holes can be drilled in perfect line, and always through the center of the bar. By the use of a stop clamped across the end of the V-block, the attachment forms a jig which can be used for a great variety of duplicate drilling.*

A jig for drilling cotter-pin holes, which facilitates the operation considerably as compared with the way it is commonly done, is shown in Fig. 41. It consists of two pieces of steel forming a clamp, each piece having a V-groove to receive different diameters of studs. The

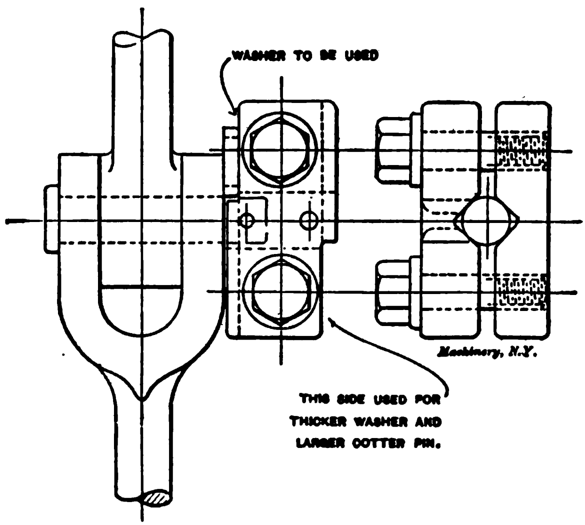


Fig. 41. Jig for Drilling Cotter-pin Holes

upper one contains two holes which correspond with the size of cotter-pins desired. Should more than the two sizes be required, extra top pieces can be used with the same bottom piece. Part of the upper piece is cut away on each side in line with the edge of the holes, which allows the washer to be used to be inserted at the recess, and the jig then clamped in position. By this means no scribing or spotting is necessary and a much better job can be done. Although it is shown so in the cut, it is obvious that the male portion of the joint need not be in position when drilling.

Jigs for Drilling Collars

The jig shown in Fig. 42 has been used with much satisfaction for drilling set-screw holes in collars. The collar *C* is held in position by means of the three locating pins *D*, *D*, *D*, and the swinging clamp *E*. In order to place a collar in position for drilling, the strap is swung

* Roy W. Harris, April, 1903.

to one side about the hand screw *G*. When the collar has been put in place the clamp is swung back, and in doing so, its motion is limited by the pin *F*, which brings it to a stop directly over the collar. At the top of the jig is a bushing *B* through which the drill is guided. When the outside diameter of the collars is likely to vary, the pins *D*, *D*, *D*, may be replaced by a central pin, *L*, as shown in the separate view in the cut, and the collar held on this while it is being drilled.*

The jig shown in Fig. 43 is designed for drilling the holes in the center of collars, and the method of drilling, described below, also suggests the value of systematizing the work in using jigs. The collars to be drilled are made of annealed tool steel in sizes varying in thickness as well as in diameter and size of hole, and are cut off from the

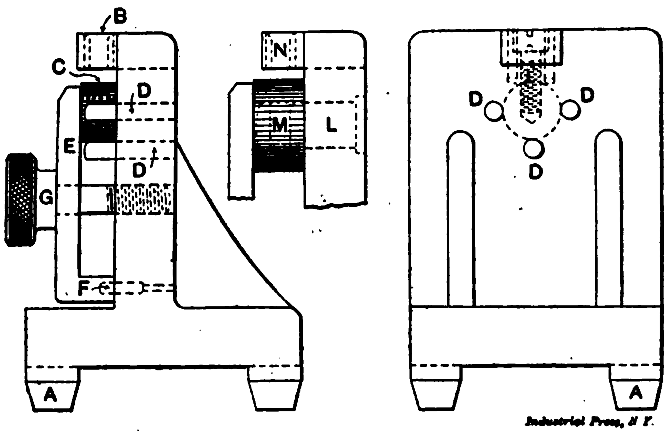


Fig. 42. Jig for Drilling Set-screw Holes in Collars

bar on the cold saw. There being a three-spindle gang drill in the shop, which was idle part of the time, it was decided to make use of it in the production of these collars. Four jigs like the one shown in the cut were made. They were made to take any diameter or thickness of collars within their range. The body of the jig is a square block of steel, with the hole to receive the collars exactly in the center. The lower end is threaded left-hand to receive the piece *B*, which has a square hole in the center to receive the wrench *C*. The ring *D* is bored taper, and fits the collar operated upon at the top end only, so that the collars will drop out of the jig easily. Different rings are made to fit collars of different diameters, and are just an easy drive fit in *A*, the body of the jig. They are driven out with a soft punch through hole *E* in piece *A*. Drill bushings *F* are also interchangeable. Piece *G* is a distance piece used when drilling thin collars in order to avoid screwing piece *B* into the jig too far. It is apparent from the cut that these pieces are made to fit the collar at one end, and beveled at the other to center in piece *B*. The reason piece *B* is threaded left-hand is as follows: If the collar operated upon should turn in

* C. H. Rowe, January, 1903.

the jig, the piece *B*, taking the thrust, would also turn, and being threaded left-hand would thereby tighten the collar in the jig. Piece *H* is a channel iron the function of which is to hold the jigs while refilling. In operation, one of the jigs is placed upon the drill press table under each spindle between flat strips *I*, which keep the jigs from turning, at the same time leaving them free to be removed for refilling. It will be seen that by having four jigs, and a three-spindle machine, by timing them so that they will finish the holes one after the other, it will give plenty of time to refill the fourth jig, and thereby, with an extra drill or two kept sharpened by the tool grinder, the

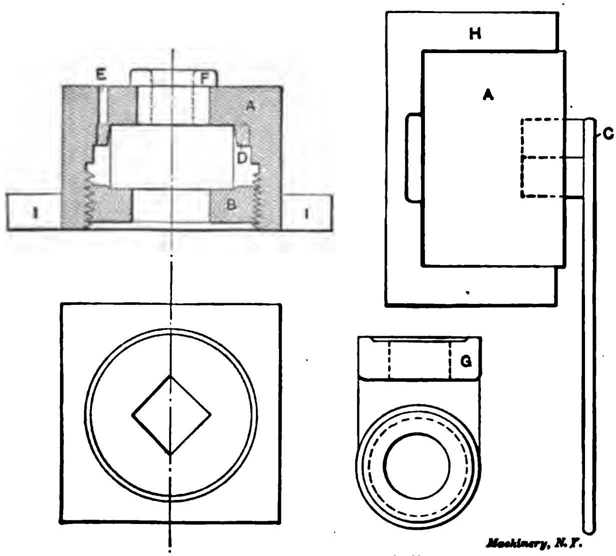


Fig. 43. Jig for Drilling Collars

machine may be kept in constant operation. This machine is equipped with a pump keeping a constant flow of cutting fluid on the drills. As this machine is also handled by comparatively cheap labor, a saving of almost 75 per cent was shown by actual test over methods previously employed in producing these collars on a turret lathe.

If it were not possible to use four jigs at a time of the kind just described, three being in operation, while one is in the hands of the operator for removing the drilled piece and inserting a new one, there would be one serious objection to the design of the jig shown. The time required for unscrewing, and again tightening, the clamping collar *B*, being threaded for its full length into body *A*, would be too long to permit rapid work. Therefore, in a case where but one jig could be used, the clamping device should be arranged so that the drilled piece can be removed, and a new one clamped in place instantly. This can be accomplished by some kind of a hinged or swinging cover, provided with a threaded binder; one half turn of the binder would be sufficient to clamp the work. In the case in hand, however, the sys-

tem of using the jigs makes this objection of less consequence, as the operator has plenty of time to attend to one jig while the collars in the others are being drilled.

Flange Drilling Jigs

Two examples of flange drill jigs are given in Figs. 44 and 45. The jig in Fig. 44 is of the simplest form for this kind of work, being merely a templet, while Fig. 45 shows the appearance and application of a more universal device, provided with an indexing plate. In cases where flanges and fittings are to be interchangeable, or to be duplicated at different times, the only accurate method of drilling such fittings is, of course, by means of a jig or templet which prevents any error arising when such parts are duplicated.

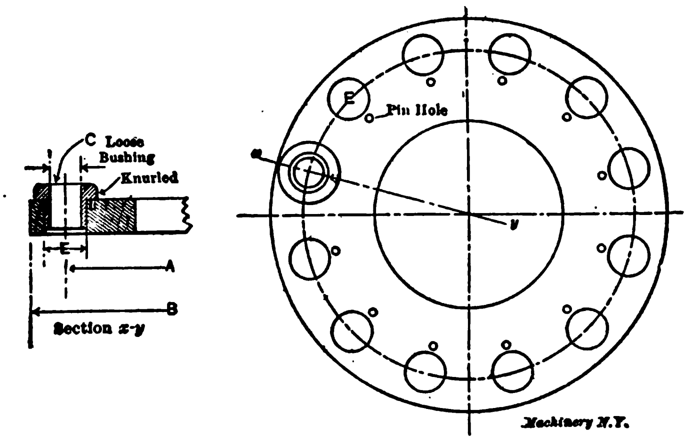


Fig. 44. Templet Jig for Drilling Flanges

The templet, Fig. 44, combines simplicity and cheapness. The ring proper may be made from a companion flange, the size for which the templet is to be used, by cutting off the head and finishing all over, the thickness being approximately one inch. Diameter *B* is made equal to the outside diameter of the flange, and *A* is the diameter of the bolt circle. A removable bushing, such as shown in section *x-y*, is used and moved from hole to hole as required. The advantage of this loose bushing over a stationary one in each hole is obvious, lessening the cost of the templet more than one-half. The bushing is made from machine steel, knurled where indicated, and hardened. The small pin prevents the bushing from revolving in its hole with the drill. In such cases where a drilling job calls for the same number of bolts in the same bolt circle, but different sizes of bolts, all that is necessary is to have two bushings, with the same diameter *E*, while *C* is made to correspond with the diameter of holes required.*

In Fig. 45 an adjustable type of jig and the work for which it is used are shown. As the number of holes in the work to be drilled, as

* Calvin B. Ross, May, 1906.

well as the diameter, varies, it would cost considerable to make individual jigs to do the work. The features of this jig are a small center plate, provided with holes for indexing, as shown at the center of the cut, and a removable arm which carries the drill bushing. The index plate is held in position by a nut on the under side of the work, and the position of the arm is fixed by a plug or pin which passes through the arm and into the plate. The bolt at the outer end of the arm is made of a suitable form to clamp on the under side of the work, and is tightened by the handle shown, which avoids the use of a wrench.

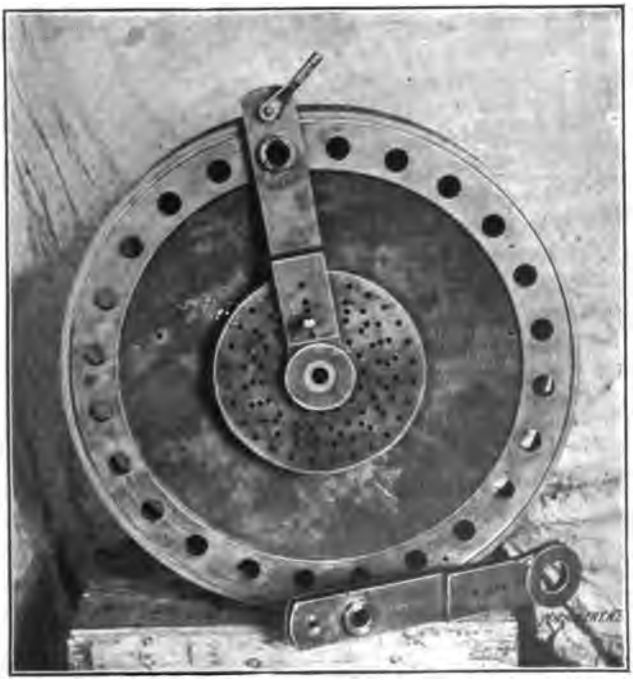


Fig. 45. Adjustable Flange Drilling Jig

By loosening this handle and withdrawing the locating plug, the arm can be turned to the next division, the plug inserted and the hole drilled. Different diameters may be drilled by using arms of suitable length, the same dividing plate answering for a wide range of sizes.*

Jigs of the description shown in the cuts, Figs. 44 and 45, are, of course, not intended for extreme accuracy, but rather for combining the objects of rapid production of work within commercial limits of accuracy, cheapness of tools, and possibility of accommodating a wide range of work with the same devices.

Adjustable Jigs

It is not always possible to provide jigs with adjustable features, particularly not when a great degree of accuracy is required. A great

* M. A. Palmer, June, 1907.

many cperations in the shop, however, permit of so wide a limit of error that fairly accurate jigs can be designed which, having a certain degree of flexibility, will accommodate a variety of work. These jigs are valuable in a double measure. In the first place they save a great deal of outlay for individual jigs, and, secondly, many a little job, for which no individual jig would be warranted, may be drilled in an adjustable jig at a great saving of time and gain in accuracy.

The jig shown in Fig. 46, in use in the W. F. & John Barnes shops, Rockford, Ill., is designed with the purpose of securing adjustability, so as to adapt the jig to pieces of different shapes and dimensions. The base piece *A* supports an upright *F*, to which the knee, *E*, is bolted. This knee holds the drill bushing and is tongued and grooved to the

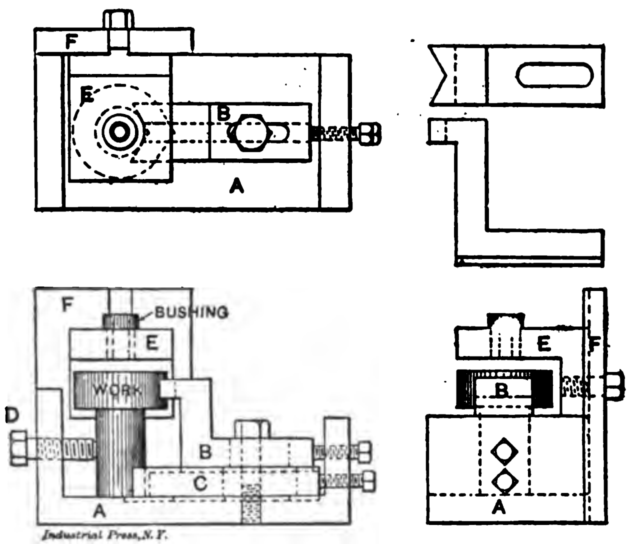


Fig. 46. Adjustable Drill Jig

upright so that it may be raised or lowered for work of different heights. The work is held by two slides, *B* and *C*, and a set-screw *D*. The lower slide, *C*, has a tongue fitting in a groove in the base, and one end is V-shaped to give support to the lower end of the work, against which it is made to bear. The slide *B* has a tongue fitting in a groove in the top of the lower slide, and may thus be adjusted independently of the latter.

An adjustable jig also provided with an indexing feature, is shown in Fig. 47. This jig is intended for drilling the clearance holes in small threading dies. As these holes are located on different distances from the center according to the diameter of the thread the die is intended to cut, one jig would be necessary for each diameter of thread in the die, although the outside dimensions of the die blanks are the same for wide ranges of diameters of thread. To overcome the necessity of so many individual jigs, an adjustable strap or slide *C* is pro-

vided, which can be adjusted to drill holes at different radii from the center of the blank, and will locate the center hole in the blank when a mark on the slide coincides with the "center line" graduation on the holder plate. The die blank is placed in holder *B*, being secured therein by the set-screws located as shown. This holder is readily rotated, as it is knurled on the edge of the flanges. It has four equally spaced locating holes, into which locating pin *D* enters.*

Miscellaneous Examples of Drill Jigs

A type of drilling jig containing features that merit the attention of the jig designer is shown in Fig. 49. One often sees expensive and complicated jigs used where one of this type would have done as well. In some shops this type has reached a high state of development, due

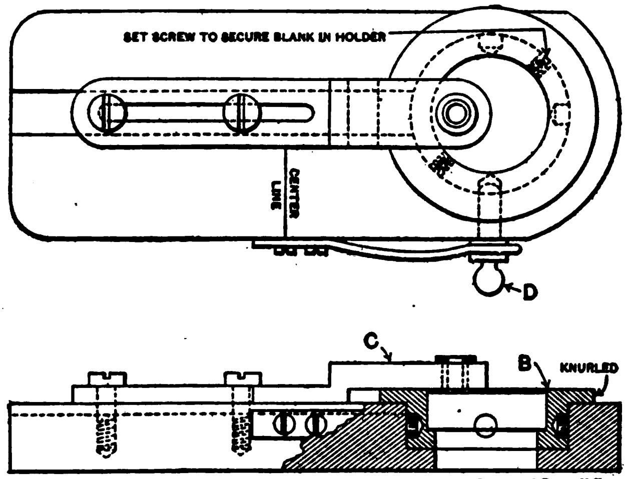


Fig. 47. Adjustable Jig for Drilling Threading Dies

to conditions that favor the adoption of a cheap and quickly made jig, namely: A constantly changing product, few pieces to be drilled of each kind, and the fact that the jigs are always wanted in a hurry. In designing jigs under these conditions, the problem resolves itself into building a cheap jig, and not accumulating a large number of useless patterns.

Fig. 48 shows the piece to be drilled and detail of clamp and feet. The plan and side views of the jig, with the work in position, are shown in Fig. 49, in which cut the jig is shown bottom side up. A cast-iron plate A is used, in which the required number of holes are drilled for the insertion of hardened bushings, and there are two locating pins, shown in the plan view at *b b*. *C* is a locating and clamping plate which is kept central by the four pins *d*. The V in the clamping plate locates the work in a central position, and as the plate also

* I. B. Niemand, February, 1902.

extends over the top of the work and clamps down upon it, it holds the work securely in place. The clamp is bolted to the plate by the screw *h*, and the work is clamped by screw *g* at the other end. Four

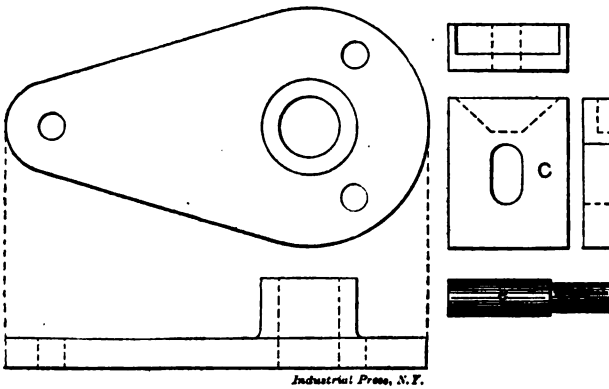


Fig. 48. Piece to be Drilled in Jig, Fig. 49, and Detail of Clamp and Feet

legs *c* support the body plate *A*, and raise it up high enough so that the work clears the table when the jig is placed in position for drilling. The oblong hole in the plate *C* permits the clamp to be moved back far enough to get the work in and out of the jig.

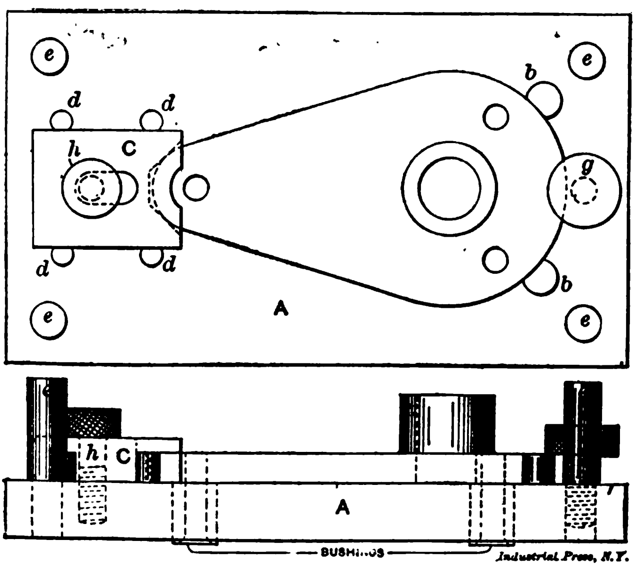


Fig. 49. Jig for Drilling Piece Shown in Fig. 48, with Work in Position

Large size plates, all planed up, may be kept in stock for the jig bodies so that pieces of the required size can be readily cut off when

needed. This jig is very accurate, as with it the work can be brought close to the plate containing the drill bushings.*

The drill jig shown in Fig. 51 has proved very efficient for maintaining uniformity in the pieces drilled in this jig, one of which is shown in Fig. 50. For the work it is intended to do, this jig is rigid and simple and is designed to withstand the severe handling that a tool

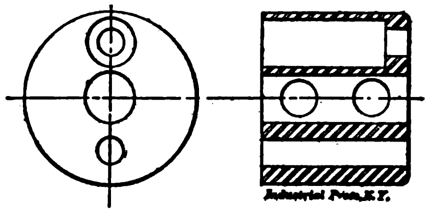


Fig. 50. Work to be Drilled in Jig, Fig. 51

of this kind usually receives from unskilled workmen. The pieces to be drilled are first turned in the lathe to the proper size and length, and the hole through the center is drilled at the same time. The object of the jig is to drill the side holes and the two end holes, which are diametrically opposite, one of them being stopped off at ⅛ inch from the bottom and continued through with a smaller size of drill.

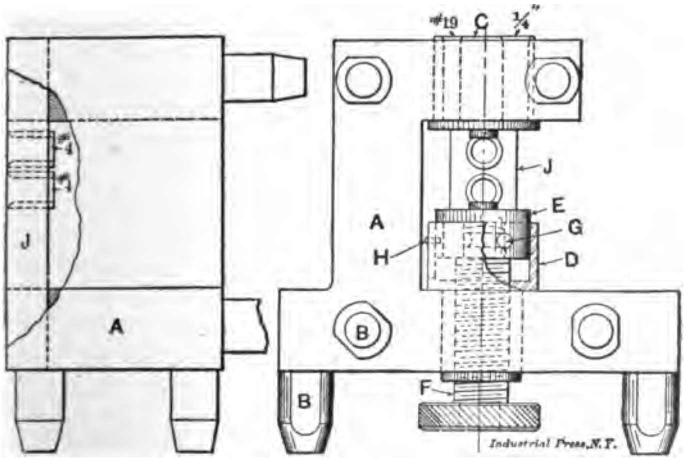


Fig. 51. Jig for Drilling Work shown in Fig. 50

The jig consists of an L-shaped casting, *A*, which is supported on its bottom and side by the steel legs, *BB*, the faces of which are hardened and lapped true. A hole is drilled straight through the jig from top to bottom, and into the top of this hole is forced the bushing, *C*, of tool steel, having a No. 19 and a ¼-inch hole, and also a guide for one end of the work projecting at the center on the lower side. The bushing

* Louis Meyers, February, 1902.

C is forced into the jig from the inside until the shoulder bears firmly against the upper arm of the jig. This combined bushing and guide is made in a single piece, instead of inserting drill bushings and a guide piece separately, because the variation allowed for the holes is greater than any that is likely to be incurred in the hardening of the bushing.

Fitted tightly in the hole in the base of the jig is the sleeve, *D*, which carries a traversing piece, *E*, with a guide point on one end directly opposite and like the one in the upper bushing. These guides fit the hole in the work, which is advanced or withdrawn by means of the screw *F*, which is fastened to the piece *E* by the pin, *G*, introduced in such a location that the side rests in a round groove on the upper end of the screw, attaching it thereto and at the same time permitting it to rotate freely. The end of a small pin, *H*, enters a spline in the side of *E* and checks any tendency to revolve when the screw is being turned. A knurled head is pinned and riveted on the

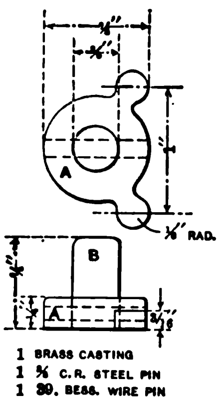


Fig. 52

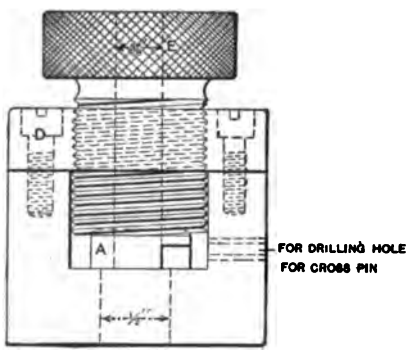


Fig. 53

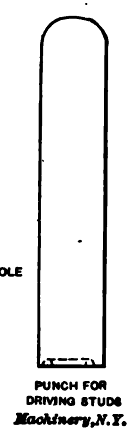


Fig. 54

Drilling and Assembling Jig

end of *F*. A strip of machine steel, *J*, of sufficient length to extend from top to bottom of the jig, is seated in a rectangular milled channel and fastened by screws and dowel pins. The side holes are carefully located in this strip, and two hardened and ground bushings for No. 4 drills are pressed in.

When in use, the work is slipped on the upper guide point, and, when the piece *E* is advanced by the hand screw, it is held firmly in place, being properly located in relation to the bushings by the center hole. The piece is then drilled as in ordinary jig drilling, the finished piece being removed by simply loosening up the hand screw. The piece *E*, with the exception of the guide on its end, is left soft for the point of the drills to enter the necessary depth for clearance.*

* C. H. Rowe, December, 1903.

Drilling and Assembling Jig

Sometimes drill jigs are designed for the performing of other operations in connection with the work than that of drilling alone. In Fig. 53 is shown a combined drilling and assembling jig which was designed and made for the purpose of facilitating the manufacture of the part shown in detail in Fig. 52. This detail consists of a brass casting *A* having a small machine steel stud *B* driven into its center and securely held against turning by a small bessemer wire pin through both casting and stud.

During the ordinary course of manufacturing with a plain drilling jig some difficulty was experienced in driving the studs squarely into the casting, thereby making it impossible to replace the pieces in their proper position in the jig in order to drill the small pin holes. To overcome this difficulty and insure the production of interchangeable work, the jig shown was designed to drill the necessary two holes before removing the part from the jig. It is very simple in construction, consisting of a cast iron body *C*, and a soft steel cover *D*, fitted with a tool steel screw bushing *E* for locating and fastening the casting in its proper position for drilling. The work is slipped in and removed from the front of the jig which is open, as shown.

To relieve the shearing strain on the small cross pin, it is necessary that the ⅜-inch hole shall be drilled a trifle small in order to make a good fit on the stud. This is accomplished by using a ⅜-inch drill that has been almost entirely used up and is therefore about 0.373 inch in diameter, thus avoiding the use of letter size or other drills that are not standard. After drilling, the jig is turned bottom side up and the stud inserted through the ½-inch hole in the bottom and driven home with the aid of the punch shown in Fig. 54. This is simply a piece of ½-inch drill rod having a groove turned at one end to clear the burr made by the drill. It is evident that when driven in this manner, the stud must go in square, and when the punch strikes the brass casting in the jig the stud has been driven to its proper depth, that is, flush with the bottom of the brass casting. The small pin hole is then drilled and the finished part removed by unscrewing the bushing.*

This jig permits rapid work, on account of its simple and efficient device for clamping the screw bushing. Clamping devices in jigs should always be designed with the object of very rapid tightening and releasing. This is particularly important in cases where only one or a few small holes are drilled in a piece, as then, if the clamping of the work consumes a rather long time, it often happens that most of the operator's time is spent in unscrewing and tightening long threaded studs or screw bushings, while the drilling operation itself takes but a trifle of the time, and the machine is, in fact, idle the greater part of the working day. Rapid insertion of work in jigs, and quick acting clamping devices, constitute one of the chief principles in jig design.

* H. J. Bachmann, December, 1905.

Simplicity in Jig Design

Another of the chief characteristics in jig design, which should be aimed at as much as possible, is simplicity. It is comparatively easy to design a complicated drill jig for almost any work, and one of the main differences between the amateur and the experienced jig designer is the latter's ability to attain, by simple means, the same results, and the same accuracy, as the former reaches by elaborate devices.

An example of a simple jig which performs the work for which it is intended as satisfactorily as a more complicated tool, is shown in Fig. 55. The work to be drilled is shown at the top in perspective. At *A* is a ⅞-inch tapped hole in a curved surface, as shown; *a a* are two

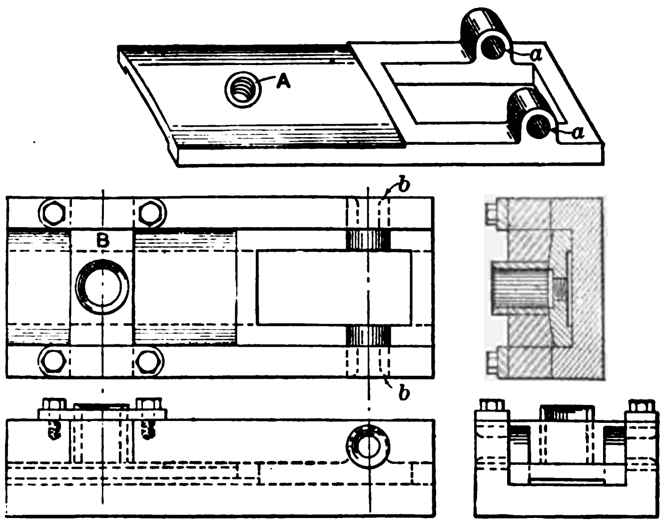


Fig. 55. Simple Design of Drill Jig

½-inch holes in the ears, which must be 7 inches from center to center from hole *A*. A cast iron jig body, of the right size to hold the piece of work inside, was made, and bushings *b b* inserted for drilling the holes in the ears. For drilling the ⅞-inch holes *A*, a cross-piece *B* was fitted into recesses cut in the sides of the jig body and this cross-piece carried a bushing, as shown. This cross-piece was held in place by two straps, as indicated. As the hole had to be countersunk, a combined drill and countersink was made, which did both operations at one cut. The work is pushed into the jig from the end, and some clamping arrangement, two C-clamps, for instance, will serve to hold it in position while drilling.*

Jigs for Drilling Rough Castings

There is a great difference in the principles of jig design applying to pieces of work which have finished surfaces from which the work may be located, and castings which are drilled directly as they come

* Frank C. Hudson, May, 1902.

from the foundry. It is not very difficult to design a jig when there is some part of the casting finished to size, but when there is practically nothing to start from, it becomes quite a different matter. If we are to judge from the number of discarded jigs in the shops, it seems that quite a few tool designers have "fallen down" on this problem.

One principal feature of these jigs is the screw bushings, two of which are shown enlarged in Fig. 58. By screwing down on the bush-

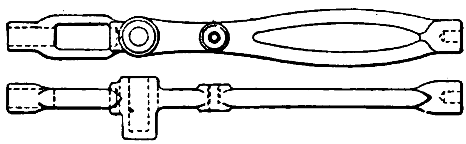


Fig. 56. Work to be Drilled in Jig, Fig. 57

ing the casting is clamped between the screw bushing and a plain bushing in the bottom of the jigs. Thus it will be seen that these bushings perform the double function of locating the hole and also holding the casting securely in its proper position in the jig. When only one end of the boss is accessible, the plain bushing cannot be used, and other means must be devised to back up the thrust of the screw bushing. Being movable, screw bushings will take care of any reasonable variation in the size of the castings and also insure that the hole shall be drilled in the center of the boss, the bushing being recessed

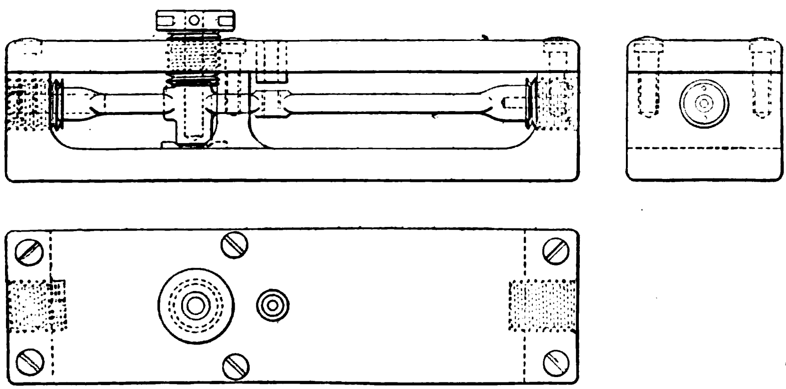


Fig. 57. Jig for Drilling Work Shown in Fig. 56

in the portion binding against the boss in order to center it. This latter condition is very desirable in work of this kind, for the sake of appearance and strength. In this form, screw bushings are rendered applicable to all forms of castings having any kind of a circular projection or boss over which the bushings may be fitted, as shown in the cuts, Figs. 57 and 60.

When headless bushings are necessary (as on both ends of the jig, Fig. 57), they are tightened down with a spanner, whereas a plain drill rod pin is sufficient for the other. When both ends of the boss are held by bushings, the holes to receive these bushings must be in

line, and when they are so aligned, it is impossible for the hole to come out of center on either end of the boss. The simplest and safest way to align these holes is to run a single-pointed boring bar through the screw bushing into the bottom of the jig, after the screw bushing has been fitted to the jig, the shank of the boring bar, of course, being a good fit in the hole of the screw bushing, which has been previously lapped to size. On the larger sizes of bushings, it has been found advantageous to use a good quality of machine steel, case-hardened and having a smaller tool steel bushing inserted in the center. When made

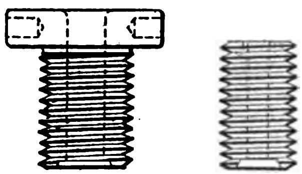


Fig. 58. Screw Bushings

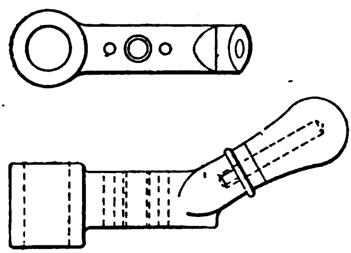


Fig. 59. Work to be Drilled in Jig, Fig. 60

entirely from tool steel, the distortion in hardening is too great to allow a good fit, which is essential on the threaded portion. The bodies of the jig should be made of cast iron, cradle-shaped, and cut out where possible, to facilitate cleaning. The covers which hold the screw bushings should be of machine steel, held in place by means of screws and dowels.

Two examples of jigs of this class are shown. The larger jig, Fig. 57, was designed for drilling the breast drill frame shown in Fig. 56.

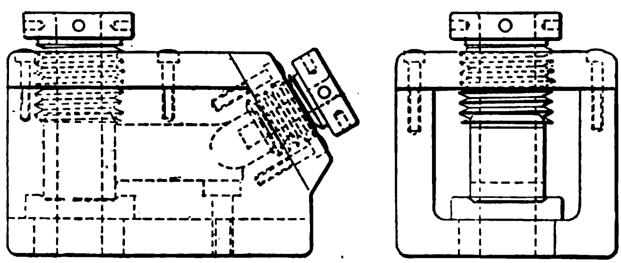


Fig. 60. Jig for Drilling Work Shown in Fig. 59

The casting is clamped by the large bushing first, and then the smaller bushings on the ends are brought up just tight enough not to cause any spring in the casting. There are two holes in this frame which must be reamed square with each other. After trying unsuccessfully to ream the holes by hand after drilling in the jig, the holes were reamed in the jig as follows: The hole in the bushing was made the exact size of the hole to be reamed in the casting. A drill of this size was used to spot the hole, following with a reamer drill, and lastly with a rose reamer, making in every respect a satisfactory job.

The difficulty with a jig of this design, in general, is that the castings will warp, throwing them out of true, and it is then not possible to locate them as described. The Hoefer Mfg. Company, Freeport, Ill., has found that in drilling pieces of this character, the stop underneath, in the center of the jig, in which the boss of the casting to be drilled rests, should be made adjustable, with a spring under the stop. The tension of this spring should be just enough to carry the weight of the piece. When placing the piece in the jig the two end bushings are then adjusted so that the boss of the work centers in the stop. By

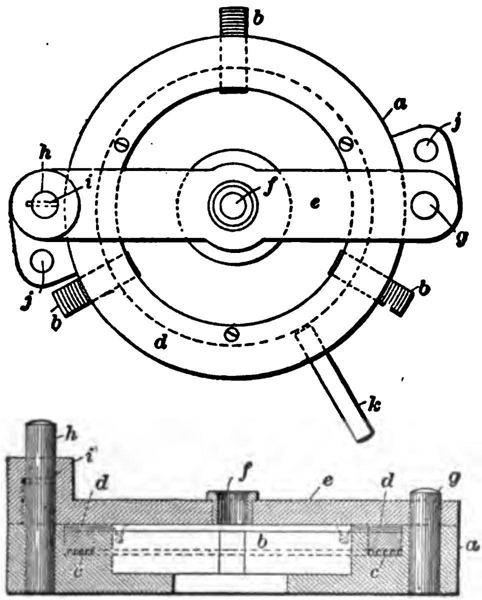


Fig. 61. Jig for Drilling and Boring Cast Gears

means of a clamping screw this stop is then held rigidly in place, and the drilled bushing finally screwed down tightly from the top.

The smaller jig, shown in Fig. 60, designed for the simple lever shown in Fig. 59, presents no difficulties beyond the drilling and tapping of the hole for the wooden handle at an angle of 30 degrees. An adjustable stud screwed into the bottom of the jig resists the pressure of the bushing on the angle. In this jig it is also necessary to clamp the larger boss first, so that when the smaller bushing is tightened, there will be no tendency to displace the casting. The same procedure was followed in the case of the tapped hole as in the case of the reamed hole in the jig previously described, namely: full size drill to spot, tap drill and then the tap itself. This latter was operated by means of a tapping attachment with friction clutch. It is hardly necessary to say that these jigs are most profitably employed in connection with a multiple-spindle drill press.*

* H. J. Bachmann, December, 1904.

Jig for Drilling and Boring Gears with Cast Teeth

Cast spur gears should be held from the outer ends of the teeth at three points as nearly equally spaced as possible. The holding should be done by jaws moving to and from the center. The outer ends of the teeth are selected because they are less liable to distortion by the washing of sand and by swelling than other parts of the teeth, and any slight lumps are much more likely to be removed in the tumbling barrel from the ends than elsewhere. Again, it is much more convenient to hold them from these points than otherwise. If three equally spaced points on the periphery of a gear are held true with the jig

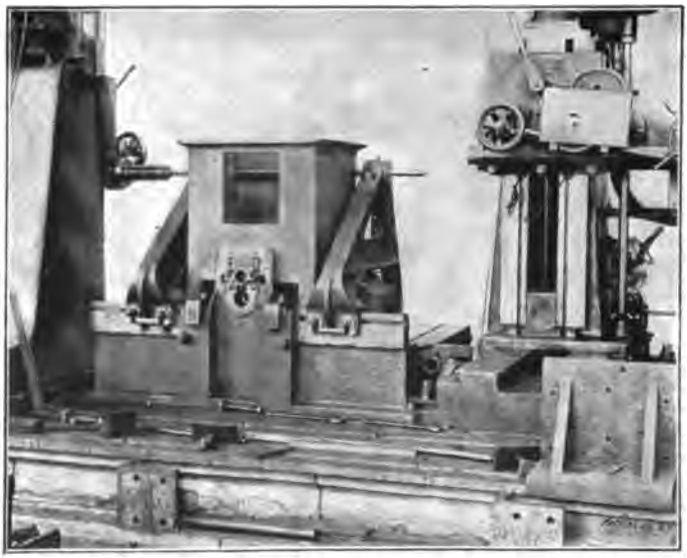


Fig. 62. Large Drilling and Boring Jig for Machine Beds

bushing, all intermediate points must be well located for the boring operation.

The manner of holding as described may be accomplished by several devices. One of these is a special form of scroll chuck. The same jig chuck may be made to accommodate different sizes of gears within a limited range with, it may be, the exchange of bushings.

Fig. 61 shows top and sectional views of such a jig. Referring to this figure, *a* is the main body casting, which is planed to receive the three steel jaws *b* and turned to admit the scroll ring which will be seen at *c*, while *d* is a steel ring used to retain the jaws and scroll. At *e* is seen the cross-bar for the support of the bushing *f*. This cross-bar is held in place by two guide pins *g* and *h*. The latter is longer than the former, so that when a gear is to be removed from the jig, the cross-bar may be raised only sufficiently to clear the short guide pin and then swung aside upon the longer one. The end of the cross-bar engaging the long pin is provided with a boss of sufficient length

to insure a parallel movement and prevent cramping. At *i* will be noticed a pin driven into the tall guide pin and left projecting into a slot. This acts as a stop to prevent the cross-bar from slipping down again when swung aside until again brought into line with the short guide pin. At *j* are seen holes for attaching the jig to the drill press table. A handle for revolving the scroll ring is shown at *k*.*

Drilling and Boring Jig for Machine Beds

The jigs shown hitherto have, in general, been intended for work of comparatively small dimensions. Modern machine manufacture, however, has developed jigs of unusual dimensions for very large pieces of work. The jig shown in Fig. 62 is used at the works of the Landis Tool Co., Waynesboro, Pa., for drilling and boring the beds of their smallest size grinding machines. The cut shows the work in progress on a large horizontal boring mill. The jig consists of a base provided with an adjustable plate for drilling the holes in the front, and adjustable brackets for guiding the bars for boring the ends of the bed. The base consists of a heavy casting, planed at the top, so as to correspond with the planed portion of the top of the bed, so that the latter may be laid bottom up on this base and located transversely by the planed lip on the front of the bed, suitable clamps being provided to hold it firmly in position. At the front of the base of the jig is a vertically projecting flange or apron of sufficient size, and so shaped as to conform to the shape required for locating most of the holes in the front of the bed; at the back part of the base is a smaller flange adapted for carrying a bushing for guiding the bar for one of the larger of these holes. Suitable T-slots are provided in the base for bolting on the various parts, and at the bottom two right-angle grooves are planed to provide for a tongue for locating on the floorplate of the boring mill. This jig is designed to accommodate two sizes of beds or similar cross sections but of different lengths, the difference being such as to only affect the location of the end brackets and some of the holes in the front of the bed. To provide for the difference of these latter holes, the adjustable plate in the front is so designed that it can be located by dowel pins in either of two positions required, and is provided with slots for clamping bolts. When boring the holes in the ends of the bed, the base of the jig is, of course, turned from the position that it has, when the front holes are bored, to the position shown in the cut. The end brackets are clamped in place, being located on the finished surface of the base. T-slots are provided so that these brackets may be shifted in or out to accommodate the different length of the beds.†

Jigs with Pneumatic Clamping Devices

During the last few years compressed air has more and more become one of the necessary adjuncts of the machine shop, and many firms employ it extensively in the operation of automatic machinery and special tools. The line cuts Figs. 63 and 65 and the half-tone Fig. 64 show a pneumatic clamp drilling jig which was designed for holding

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* Cyril B. Clark, June, 1904.
† H. F. Noyes, March, 1907.

small castings, pinions, spur gears, sprockets, pulleys, etc., for reaming or drilling. This type of jig is used with great success in one of the largest manufacturing concerns in Chicago. Formerly castings of the nature named were held in a jig, using a screw bushing mounted in a swinging arm to hold the work while drilling; the arm was swung around over the casting and the bushing was screwed down onto the work. Frequently the operator would neglect to screw the bushing

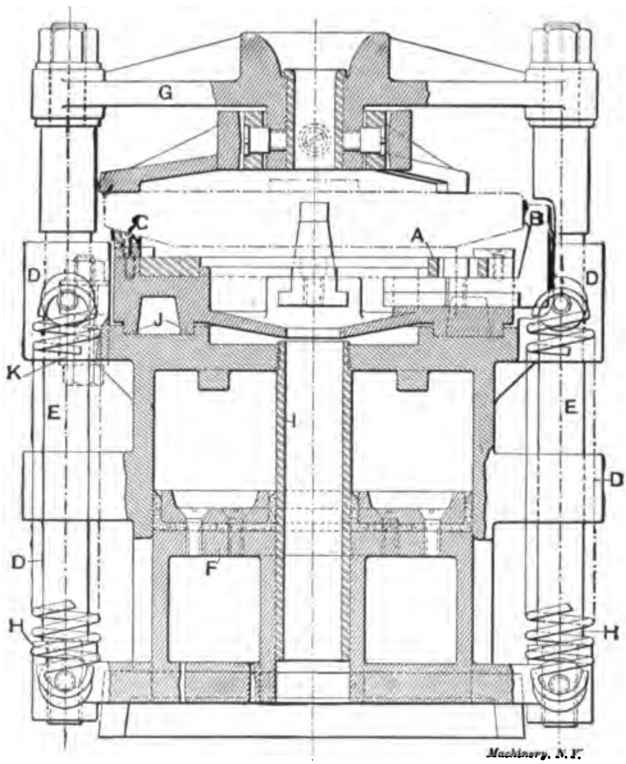


Fig. 63. Vertical Section of Pneumatic Clamping Jig

down tightly against the work, with the resultant of a bad job of drilling and a spoiled piece. In any case there was considerable time lost in operating the jig.

The air clamping drilling jig shown in section in Fig. 63 was designed to decrease the time required to operate the jig and to improve the character of the work done. The cut shows how a bevel gear is held. The gear rests on the inclined face *C*, and between three chuck jaws. Beneath the casting is a ring, *A*, having three cam eccentric slots which move the jaws *B* toward or away from the center when the ring is turned by a suitable handle. With this jig the operator needs only to turn an air valve handle to hold the work securely and

in the central position. To hold spur gears, a centering piece is used, similar to the one shown for bevel gears in Fig. 63, with the exception that the surface *C* is made flat, the jaws then being used alone to center the work.

The jig includes a cylinder having two lugs or ears *D*, which encircle the guides *E*. These guides connect the piston *F* with the cross-arm or yoke *G*, which holds the drill bushing. The admission of air to the

Fig. 64. Pneumatic Clamping Jig used for Drilling Sprocket Wheel

cylinder forces the yoke and bushing down on the work and holds it there until the piece is finished. The air is then released and the tension springs *H*, of which four are provided, pull the piston and the connected cross-arm up and release the work. Compressed air is admitted in the side of the cylinder through a pipe in which is fitted an ordinary three-way valve. The pipe *I* in the center of the cylinder is an important feature, as it permits chips to fall through the jig at the bottom instead of collecting on the top. What few chips accumulate on the top are removed by a hose leading from the exhaust port of the valve and directed against the top of the cylinder, thereby blowing the chips away with each exhaust. The centering device is made different, of course, for different pieces, Fig. 63 showing one for a "flat" bevel gear; and each pattern of pinion, gear or sprocket has to have a corresponding piece *C*. The cylinder has an annular groove *J* turned in the top and made concentric with the axis of the cylinder and of the drill jig. The centering device has two projections which fit the cylin-

der top and groove. This makes the air cylinder conveniently interchangeable with any number of centering devices, the centering device being removed quickly so that there is little time lost in making changes, the clamping being a simple matter. The cylinder has three lugs *K* with open slots for bolts, these matching with three lugs on the centering device and constituting the clamping arrangement for the centering piece. When the centering piece is to be changed, the three bolts are loosened, slipped out of the slots, and the centering piece is lifted out and exchanged for another.

If the drill bushing has to be changed, the yoke *G* is taken off and replaced by another, for it is generally desirable to have a yoke with

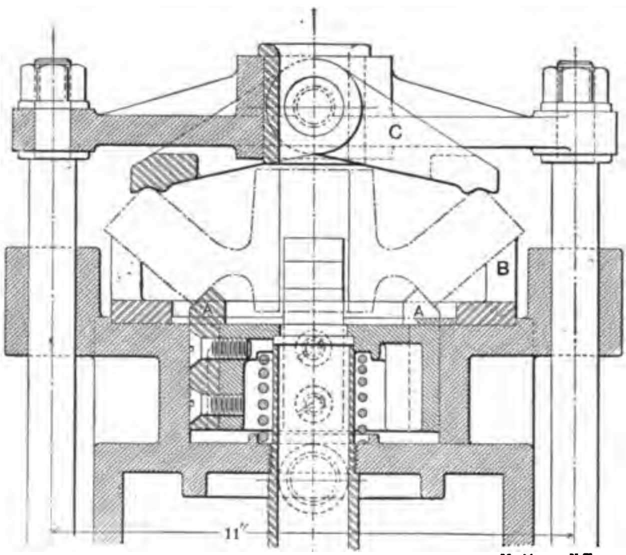


Fig. 65. Vertical Section of Pneumatic Jig Fitted with Equalizing Centering Device

its own bushing for each job. With small work the yoke simply has a bushing driven from the bottom, as illustrated in the half-tone Fig. 64, and the bushing alone presses against the work, but for larger work, which should be held down at three places on the rim, the yoke and clamp are connected with a universal joint as illustrated in Fig. 63, thus insuring equal pressure on the three clamping points.

Fig. 65 is a centering device, used on the air-cylinder, in which there is a float. This float rests on a heavy spring, and on the float are three lugs *A* which support the gear casting. This device centers the casting, while the yoke is pulled down by air pressure until the gear rests on the three stationary surfaces *B*. The yoke with its equalizing saddle *C* holds the bevel gear firmly while drilling.*

In the design of all devices using compressed air, care should be taken to economize as much as possible with the air, making the spaces

* O. C. Bornholt, April, 1907.

it has to fill as small as possible. In the jig shown this has not been thoroughly taken into consideration. The long motion of the piston, entirely operated by air, makes it necessary to fill a great space with air each time the work is clamped. In such cases it is usually possible to move the clamp down upon the work with a mechanical movement requiring no air, and then effect only the actual clamping by the compressed air, in which case it would probably not be necessary to use one-tenth the amount of air now used in the jig

CHAPTER IV

DIMENSIONS OF STANDARD JIG BUSHINGS

In the design of drill jigs there is little save experience and judgment to guide the draftsman when determining the dimensions of the drill bushings. This often results in having bushings for the same size of drill made with widely varying dimensions as to length and outside diameters. If, on the other hand, some standard is adopted

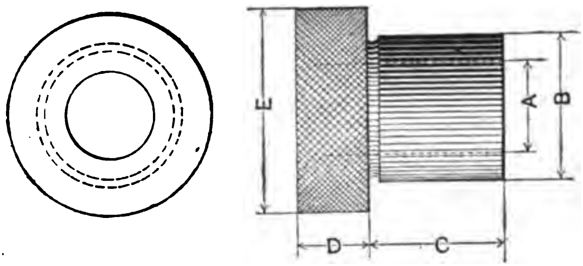


TABLE 1. DIMENSIONS FOR STANDARD FIXED JIG BUSHINGS

Size Drill. A	B	C Short.	C Med.	C Long.	D	E
60	1/4	1/4	3/8	5/8	3/32	5/16
3/16	3/8	1/4	3/8	5/8	3/32	7/16
1/4	1/2	1/4	3/8	5/8	3/32	5/8
3/8	5/8	3/8	1/2	5/8	1/8	3/4
1/2	3/4	3/8	5/8	7/8	1/8	7/8
9/16	13/16	1/2	3/4	1 1/8	1/8	1
5/8	7/8	1/2	3/4	1 1/8	5/32	1 1/16
3/4	1	1/2	3/4	1 1/4	5/32	1 3/16
7/8	1 1/8	1/2	3/4	1 1/8	3/16	1 5/16
15/16	1 3/16	5/8	7/8	1 1/4	3/16	1 7/16
1	1 1/4	5/8	7/8	1 1/4	1/4	1 1/2
1 1/16	1 5/16	5/8	7/8	1 3/8	1/4	1 9/16
1 1/8	1 7/16	5/8	7/8	1 3/8	1/4	1 3/4
1 1/4	1 5/8	5/8	7/8	1 3/8	1/4	1 15/16

and adhered to, uniformity will be insured and the toolmaker can make up bushings in leisure moments, knowing that they will be available whenever a rush job of jig work comes along. Tables 1 and 2, which give dimensions of bushings, are now used by a large manufacturing concern, and furnish an excellent guide for any draftsman designing jigs where no standard has been adopted.

Table 1 gives dimensions for bushings which are to remain in the jigs permanently, and in making these bushings the external diameter given in the column *B* would be made a driving fit in the hole in

the jig. Table 2 gives dimensions for removable bushings, and in this case the outside diameter would be made a light sliding fit in the hole. In both tables the column *A* indicates the size of drill for which the

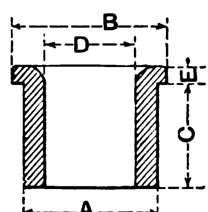
TABLE 2. DIMENSIONS FOR STANDARD REMOVABLE JIG BUSHINGS

Size Drill. A	B	C Short.	C Med.	C Long.	D	E
60 to 40	5/16	3/8	1/2	3/4	1/4	9/16
40 to 26	7/16	1/2	5/8	3/4	5/16	3/4
26 to 1	7/16	1/2	5/8	3/4	5/16	3/4
7/32 to 19/64	1/2	1/2	5/8	1	3/8	15/16
5/16 to 3/8	5/8	5/8	3/4	1 1/4	3/8	1
25/64 to 7/16	11/16	5/8	7/8	1 1/4	7/16	1 1/8
29/64 to 17/32	3/4	5/8	7/8	1 1/4	7/16	1 1/8
35/64 to 39/64	7/8	5/8	7/8	1 1/4	7/16	1 1/4
5/8 to 11/16	15/16	3/4	1	1 1/4	1/2	1 3/8
45/64 to 49/64	1	3/4	1	1 1/4	1/2	1 7/16
25/32 to 7/8	1 1/16	3/4	1	1 1/4	1/2	1 1/2
57/64 to 15/16	1 3/16	3/4	1	1 1/4	1/2	1 5/8
15/16 to 1	1 1/2	3/4	1	1 1/2	1/2	1 3/4

bushing is to be used, and the hole in the bushing would be made from 0.001 to 0.003 inch larger than nominal size. The bushing shown in the cut above Table 1 has a knurled head. Of course, the head is only knurled on removable bushings.*

TABLE 3

BUSHINGS FOR DRILLING AND REAMING JIGS									
D	A	B	C	E	D	A	B	C	E
3/16	7/16	9/16	5/8	3/32	1 3/8	2 1/4	2 1/2	1 1/2	1/4
1/4	1/2	5/8	5/8	3/32	1 1/2	2 7/16	2 11/16	1 1/2	1/4
5/16	9/16	11/16	5/8	3/32	1 5/8	2 5/8	2 7/8	1 1/2	1/4
3/8	11/16	13/16	5/8	1/8	2	2 3/4	3	1 1/2	5/16
7/16	3/4	7/8	5/8	1/8	2 1/4	3 1/8	3 3/8	1 1/2	5/16
1/2	13/16	15/16	3/4	1/8	2 1/2	3 3/8	3 5/8	1 1/2	5/16
9/16	7/8	1 1/8	3/4	1/8	2 3/4	3 5/8	3 7/8	1 1/2	5/16
5/8	1	1 3/16	3/4	1/8	3	4	4 3/8	1 1/2	5/16
11/16	1 1/16	1 1/4	3/4	1/8					
3/4	1 1/8	1 3/8	1	3/16					
13/16	1 3/16	1 7/16	1	3/16					
7/8	1 5/16	1 9/16	1	3/16					
15/16	1 3/8	1 9/16	1	3/16					
1	1 7/16	1 11/16	1 1/8	3/16					
1 1/8	1 5/8	1 7/8	1 1/8	3/16					
1 1/4	1 3/4	2	1 1/4	3/16					
1 3/8	1 15/16	2 3/16	1 1/4	3/16					
1 1/2	2 1/8	2 3/8	1 1/2	1/4					



Machinery, N.Y.

Table 3 gives the dimensions for drill bushings for a wider range of drill sizes, according to the adopted standard of a large manufactur-

* MACHINERY, April, 1904.

ing concern in Chicago. It will be noticed that the shoulders are much smaller than generally used. There is no real need for the shoulder of a loose or removable bushing to be larger than is necessary for a good finger hold. By keeping the shoulder dimensions down to the figures given in the table, a considerable saving of steel is effected in the larger sizes, and when this amount is multiplied by the thousands of bushings necessary in large machine shops, it becomes a very important matter. Another feature of economy possible in bushings is the use of machine steel, case-hardened, which gives as good results for some work as tool steel, and of course is far less costly, both in price per pound and in time required for working.*

Hardening Small Jig Bushings

To harden large quantities of small jig bushings without danger of cracking under the head while hardening or while driving them home, proceed as follows: Put one gallon of fish oil in a suitable metal bucket, and place this in a larger bucket of cold water. The bushings, strung about six on a wire, are heated in a small blow torch fire to a light red heat and are then quickly plunged into the oil, and kept moving around until cold. The hardness will depend upon the degree of heat given, and this can be so regulated that it will not be necessary to polish and draw bushings after hardening.†

* O. C. Bornholt, May, 1905.
† H. J. Bachmann, November, 1905.

CHAPTER V

USING JIGS TO BEST ADVANTAGE*

It may be deemed proper, in the closing chapter, to review, in general outlines, the principles of jig design, and to give some directions for getting the full value out of jigs.

Competition and the growing demand for machinery have necessitated the introduction of improved tools to reduce the cost. Jig and fixture designing has come to be a trade by itself; undoubtedly there is no branch of the mechanical business which requires so much practical experience as this particular line. A poorly designed tool is a very costly thing; hundreds of dollars can be wasted in a short time with an inferior one. On its accuracy, simplicity and quickness depend quality and quantity, hence cost of product.

There are a number of obstacles to be overcome in accurate jig and fixture designing. The clamping must be done quickly and without springing the jig or the work; then provision must be made for easy cleaning out of chips, and another very important thing is, that it must be so constructed that it will be impossible to get the work in the wrong way. It is important to make drilling jigs as light as possible. To obtain lightness, just as little metal must be used as is necessary to sustain the strain brought to bear upon the part. All metal should be so placed as to be in line with the strains exerted thereon; therefore, jigs should be box-shaped. The advantages obtained are manifold, for, while they are light, they are also easily cleaned. Some of the older manufacturers still advocate the use of heavy drilling jigs—large, cumbersome things, and slow to handle. Their reason is that a light jig will not stand the rough handling. While that is true in a way, there ought not to be any necessity for such rough usage. A proper system in the shop would overcome this.

It is customary in a good many of the large shops in the Eastern States particularly to hire green men and boys to operate the jigs and fixtures. If it is a drilling jig, especially a small one, the gang drill is set up for that purpose; each spindle in rotation is set up for its respective operation. The men that set these machines are competent machinists, and they always keep one or more machines set up for the first one who gets out of a job. They are also responsible for the quality and quantity of work turned out. For instance, a drill or reamer may get roughed up and in this manner spoil the work or a drill bushing. Therefore, it keeps the machinists in charge on a constant outlook. The operators are provided with a gage and a sample piece which is correct. They are instructed how to use it; also to try every few pieces to see that they are coming like the sample. In this manner one good man can direct the work of a dozen cheap ones.

* MACHINERY, August, 1904, and February, 1905.

In the following outline of a system for getting the most out of the tools in the shop, the word "jig" will be meant to include all jigs, templets, fixtures, appliances, etc., which aid in the rapid and accurate machining of parts. With such assumptions allowed, the necessity for some systematic scheme of management for the use and care of the jigs should be apparent. However, it is not uncommon, even in these days, when the jig is admittedly one of the main factors instrumental in developing the shops of the past (where machinery was "built"), into the shops of the present (where it is "manufactured"), to find concerns where the jigs are given no consideration beyond designing them and keeping them in a questionable state of repair. The whole tool or jig scheme, however, is so interwoven with the entire shop that the success of a system cannot be dependent entirely upon any one person, but upon the co-operation of all.

DRILLING JIG SET-F 42 C.
1 JIG
1 " LID
4 THUMB NUTS
2 SET SCREWS
6 BUSHES
TWIST DRILLS - 31/32" - 3/4" - 17/32" - 1/2"
TAP - 5/8" - 10 THREAD MACHINE
REAMER 1" MACHINE
" TAPER - SPECIAL NO. F 39

Fig. 66. List of Parts of Jig, and Tools used with same

DRILLING OPERATION SHEET F 42 C.
DRILL 31/32"
REAM 1"
REVERSE LID AND
DRILL 17/32"
TAP 5/8" - 10 THREADS
DRILL 3/4"
" 1/2"
REAM 1/2" SPEC. NO. F 39
NOTE:- CARE MUST BE TAKEN THAT CHIPS DO
NOT ACCUMULATE IN CORNERS OF JIG.
NOTE:- DO NOT TIGHTEN TOO MUCH ON SET
SCREWS AS THERE IS DANGER OF SPRINGING
WORK.

Fig. 67. List of Operations to be performed

The tool foreman is the one, after the management, who contributes most to either success or failure, and therefore his selection should be made with care. This tool foreman, as we prefer to call him, is to the modern shop what the head toolmaker was to the old-time shop, and his increased duties and responsibilities entitle him to the new title. He should possess executive as well as mechanical ability, and be broad-minded and up-to-date, for to him should be intrusted the tooling of the machines, the design, manufacture and care of the jigs, the complete control of the tool-room and the enforcement of any system the management may inaugurate. He will, however, be doomed to only partial success, if not absolute failure, without a tool-room system.

Suitable methods should prevail in the tool-room, or better, in the jig-room, whereby a workman when receiving a jig gets all the necessary tools to perform all the operations upon the piece that the jig is designed to do. It should not be necessary for him to ask for the tools separately, but simply to ask for the jig and tools for such or such an operation, designated either by name or number—preferably by number—and have them delivered to him complete. By doing this, much time will be saved, and mistakes will often be avoided. This can

be accomplished by giving each jig all the loose pieces belonging to the jig and all the special tools, the same number as the piece they are used upon. They should be indexed under this number and kept in suitably grouped compartments and the compartments conspicuously numbered so that they can be easily located. In these compartments is also kept a list, Fig. 66, showing what constitutes a complete set. When a jig is called for, reference is made to the index, if necessary, the compartment found and the complete set of jig and tools delivered with reference to the list.

Probably one-half of all jigs are designed to perform two or more operations, and when such is the case, to economize in time and often to obtain the best results in machining, each jig should have its operation sheet, Fig. 67. To illustrate why it is necessary to perform the several operations in a prearranged order, take, for instance, two holes intersecting at acute angles, such as a shaft hole and a locking rod hole, where the locking rod hole drills half out into the shaft hole. Ordinarily a workman would drill the larger or shaft hole first, and the locking rod hole afterward. This would be wrong, however, for the locking rod hole drill upon entering the shaft hole and meeting no resistance for half its diameter, would run out, and the hole would not be straight. A very handy arrangement is to have the tool sheet, Fig. 66, and the operation sheet, Fig. 67, mounted upon opposite sides of a cardboard. They should be of some convenient size, to be determined by the number of separate items it is necessary to put upon them.

It is regrettably too generally the custom to take for granted that a piece is right if it has been jigged, and in this way much work is often spoiled that could be avoided by the simple system of inspecting the first piece of every lot done in a jig and ascertaining its correctness. If the first piece is found to be correct, it is reasonably safe to assume that the rest will be. It is also well to provide printed blanks upon which defects and possible improvements in jigs are reported to the tool foreman. These are made out in duplicate by the foreman under whom the defects, etc., are discovered, he keeping the copy and sending the original to the tool-room. This method will be found to be superior to giving verbal instructions, as it is a check from one foreman to another. There is an adage which cannot be more appropriately applied than in the case of repairing jigs, and that is, "Don't put off until to-morrow what can be done to-day."

It seems hardly necessary to mention the matter of allowing repairs to be made upon jigs in any other place than the tool-room, because it is so obviously wrong that every one must see the fallacy of such a course and what a demoralized state of affairs it will lead to. In this matter there should be absolutely no margin. Whenever repairs are necessary on jigs, they should be turned over directly to the tool-room, and even the most trivial matters should be attended to by the man in charge of the jigs, as he is held responsible for results.

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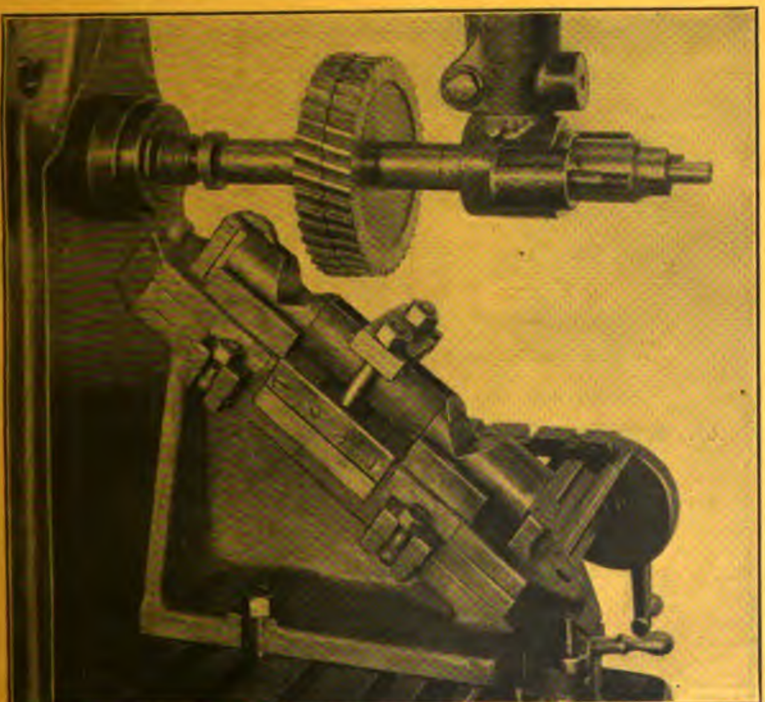

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NUMBER 4

MILLING FIXTURES

THIRD REVISED EDITION

CONTENTS

CHAPTER I

ELEMENTARY PRINCIPLES OF MILLING MACHINE FIXTURES*

The principal consideration, when designing fixtures that are to be fastened solidly to the table of a milling machine, should be to have the fixture firm enough to admit working the machine and cutter to their limit of endurance. In fact, the fixture should be stronger than the machine itself, and able to resist any possible strain that the cutter can exert. While fixtures should be strong, the movable parts should be so made as to be easily manipulated. All bearing and locating points should be accessible to facilitate the removal of chips and dirt. The action of the clamping devices should be rapid, so that no time is lost in manipulating them.

The Milling Machine Vise—False Vise Jaws

The first fixture to consider is the milling machine vise, which has a stationary and a movable jaw, against which are placed removable jaws, held in place by means of screws. The stationary-removable jaw generally has connected with it any shelf, pins, or means for locating the pieces to be machined. The reason for attaching them to this jaw is that this portion of the vise is not movable, and is, or should be, stiff enough to resist without springing any pressure that may be exerted by means of the crank and screw. The jaw attached to the movable slide part of the vise, on the contrary, is liable to alter its location slightly under strain, especially when the vise becomes worn.

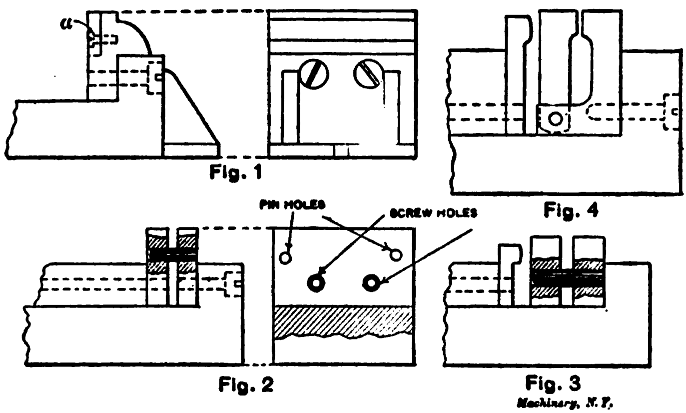
For some purposes, where but a few pieces are to be milled, or where the character of the pieces is such that there is not much liability of the jaws wearing, and thus affecting the accuracy of the pieces, it is safe to make the jaws of cast iron. If, however, there is a considerable strain on the jaws, it is advisable to make them of steel and harden them. For most purposes, jaws made of a good grade of machinery steel and properly case-hardened answer as well as those made from tool steel, and cost only a fraction as much for stock.

If possible, the piece to be machined should be held in the jaws below the level of the top of the vise in order to avoid springing the jaws out of a vertical position, as would be the case if the piece were above the level of the top of the vise. Occasionally pieces are so shaped, however, that they have to project considerably above the top of the vise jaws, in which case the jaws may be made with a rib which extends over the top of the vise and rests on the piece, as shown in Fig. 1. This furnishes a brace and prevents the springing that would prove harmful to almost any piece of work that it would be safe to

* MACHINERY, November, 1905; December, 1905; January, 1906; and February, 1906.

hold in a vise while milling. As it would prove quite expensive if many jaws of this style were made from steel, they may be made from cast iron, and a plate of steel placed where the work is to rest, as shown at *a*, Fig. 1. After the steel plate has been cut to shape and the locating device attached, the jaw may be hardened. If the devices mentioned are pieces which must be attached to the jaw, or pins which enter holes in it, they must be removed when the jaw is hardened.

At times it is necessary to hold pieces so that they rest on shelves on each jaw, or are located by pins in both the stationary and movable jaw. Generally speaking, it is advisable to construct special fixtures for such pieces, provided the degree of accuracy and the number of pieces warrant the outlay. However, if the pieces must be held in



Figs. 1 to 4. Special Jaws for Milling Machine Vise

jaws in the vise, some method should be found to prevent the movable jaw from rising when pressure is applied, in the operation of "tightening up." If the jaws are reasonably thick, large pins may be used, one near each end of the jaw, as shown in Fig. 2. These pins must be forced solidly into one jaw and fit closely in the other. Another method which works nicely is shown in Fig. 3. In this case the movable jaw proper is connected with the stationary jaw by means of pins, or a slide of different design. It is not, however, attached to the movable slide of the vise, but a hardened piece of steel is attached to this and bears against the movable part of the jaw. Many other forms are made, one of which is shown in Fig. 4. The front portion hinges at the bottom, and is pressed against the work by a movable slide. In all such holding devices, however, chips are liable to get between the various parts, decreasing their accuracy.

When making any form of holding device, it is necessary to provide a place for the burrs that are a result of previous operations, unless they are removed by a process of filing or grinding. In many cases these burrs will be removed by future operations if it is possible to provide a place for them so that they will in no way affect the accuracy

of the piece. For this reason milling machine jaws and other fixtures are cut away or recessed in places to allow the burrs a place in which to drop, as shown in Fig. 5 at *A*. At *B* a piece of work is shown with the burr mentioned.

Provisions for Removing Chips

It is the custom in most shops to provide a liberal supply of oil, or other lubricant, for cutters when milling work that requires lubrication. In many cases this fluid is used to wash out the jaws or fixtures after removing a piece of work. As this supply is used over and over, however, it is liable to become thick and gummy, and apt to prove harmful rather than helpful, unless the operator watches his fixtures closely. In some shops compressed air is used to blow chips from the working surface, and in many cases "works like a charm." On certain jobs nothing seems so effective as the hand and finger method for cleaning the surfaces of the fixtures.

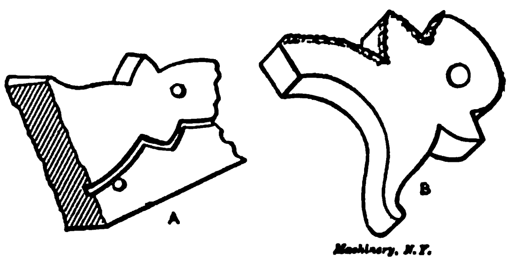


Fig. 5. Arrangement of Fixture or Vise Jaw to Accommodate Burr

One example of the necessity of taking account of the question of chips, taken from actual practice, may give this matter its full emphasis. This example also shows how at times it is necessary to change existing methods in order to accomplish the desired result. A piece of work consisting of a flange, as shown in Fig. 6, was provided with projecting portions, *a a*, which were to be straddle-milled. The jaws of the vise used to hold this piece had circular grooves, *b b*, Fig. 7, which were thought necessary to properly hold the piece, since the pull of the cutters was in an upward direction; but these grooves made an excellent place for a deposit of chips, and as it was a difficult matter to clean them, and as the operator was working by the piece at a rather low rate, and consequently was not inclined to take too great precautions, the edges of the flanges of the piece being milled became badly scored, and required an extra operation in the turret lathe to remove the marks. To overcome this difficulty, the projecting lips of the vise jaws were cut away and the direction of rotation of the cutters reversed, the overhead belt being changed so that the cutters would run onto the work, thus holding the work securely down on the seating surface of the jaws.

Special Forms of Vise Jaws

It sometimes happens that the opening in the vise is not sufficient to take in a long piece of work, in which case the jaws may be made

of a form shown in Fig. 8. At other times the vise may be used with the movable jaw of the original form, and with the stationary jaw arranged as in Fig. 9. In this case a flat piece of steel is attached to the outside of the jaw by means of screws which are a snug fit in holes drilled and reamed in both the auxiliary and stationary jaws of the vise. It is apparent that such an arrangement does not allow great accuracy, as the jaw on the end has no backing, and consequently will easily spring, yet there are instances where it answers the purpose as well as a costly fixture. If milling machine vises are drilled for screws that hold jaws in such a manner that the jaws will readily go on any vise, much valuable time may be saved. If we are equip-

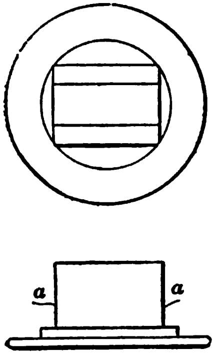


Fig. 6. A Difficult Straddle Milling Job

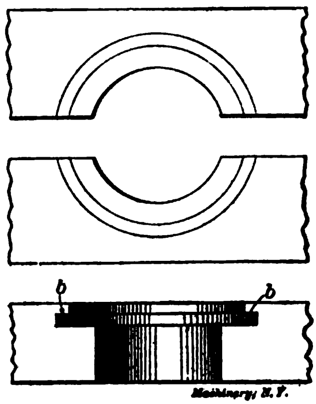


Fig. 7. Arrangement of Vise Jaws to Hold Piece Shown in Fig. 6

ping a shop with new machines, this may be readily accomplished, as we may order vises drilled alike and corresponding with some vise already in use, and to which a number of pairs of jaws are fitted.

Cams or Eccentrics for Binding Vise Jaws

The vises ordinarily furnished with milling machines are opened and closed by means of a screw. Unless it is necessary to apply considerable pressure to the piece being held, this form of vise will not work as quickly as desirable where cheapness of production is a factor. To overcome this objection, vises are made so that the slide may be opened and closed by means of a cam and lever, and unless there is much variation in the sizes of pieces being machined, the cam will cause the work to be held sufficiently firm. The work may be placed in and taken out in this way much more quickly than when a vise operated by a screw is used. In fact, where such a vise will answer the purpose, it will be found as cheap to operate and as satisfactory in results as special fixtures; and the jaws necessary when starting a new job are, as a rule, much cheaper than special fixtures.

When it is necessary to cut in the vise jaws the shape of the piece to be milled, it may be done by milling with the mills to be used on the work, as shown in Fig. 10. The pins, or other appliances for hold-

ing the work, should be added, and are then again removed and the jaws hardened.

Hardening Vise Jaws

While, as mentioned, such parts of tools as milling machine jaws are ordinarily made from machinery steel, open-hearth steel which does not contain over 25 or 30 points carbon is to be preferred. This may be case-hardened nicely in oil with little or no liability of springing, as the depth of hardness necessary does not call for extreme heat, which causes the steel to go out of shape and also opens the grain of the steel and renders it more liable to become indented should a chip be pressed against the surface. The jaws, if made of this kind of steel, may be packed in the hardening box with a mixture of charred bone and wood charcoal—equal quantities—and run five or six hours after they are red hot. Then they may be removed and dipped in a bath of oil, working them up and down lengthwise in the oil until the red has entirely disappeared, after which they may be lowered to the bottom of the tank and allowed to remain until cold.

If for any reason it is necessary to harden the piece deeper than can be done in the length of time mentioned above, then the red-hot

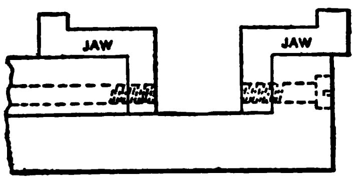


Fig. 8. Off-set Vise Jaws

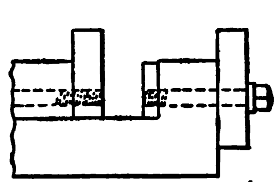


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Fig. 9. Extended Vise Jaw

jaws may be exposed to the action of the carbonaceous material for a greater length of time. If the jaws are made from the grade of stock mentioned, and are given a low heat, there should be no springing during the hardening process.

Regular Milling Fixtures

There are many pieces of work that can be machined at a much less cost if a fixture specially designed for the purpose is used. When the number of pieces to be done warrants the outlay, it is generally advisable to pursue this policy. There are other pieces of work that must be held in specially designed fixtures in order to produce a sufficient degree of accuracy, and there are still others that cannot be machined unless such fixtures are provided. The design of such fixtures should always depend on the number of pieces to be machined, and the cost of doing the work. If a fixture is to be used for machining a relatively small number of pieces, then it should be made at as small a cost as possible. If, on the contrary, it is to be used as a permanent fixture for machining the same class of work for an indefinite period, then it should be made in a manner to insure its "standing the racket." Such fixtures should be made very strong and solid, as the cost for stock and labor is not much greater than when making a too light, more or less useless contrivance.

As cast iron is the material used for the base of most fixtures of this kind, plenty of the material rightly distributed will insure freedom from chattering and uniformity of the product, provided other conditions are right. This additional weight of cast iron does not materially add to the cost of the fixture. As a rule, cast iron does not prove satisfactory as a surface against which to bed small pieces when

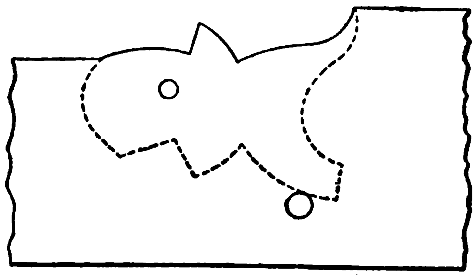

Fig. 10. Vise Jaw made to Correspond to Shape of Work

milling, and for this reason a surface of steel is generally provided for this purpose.

Examples from Actual Experience

Fig. 11 shows a milling machine fixture used for milling a leaf for a vernier rifle sight. It is necessary to have the sides, *a a*, of the leaf parallel to the sides of the slot. The base, *b*, of the fixture is made of cast iron, the bottom of which is planed flat. It has a slot cut in it to receive the tongue pieces which fit the tongue slot in the table. A groove is cut in the top surface to receive a tongue on the steel por-

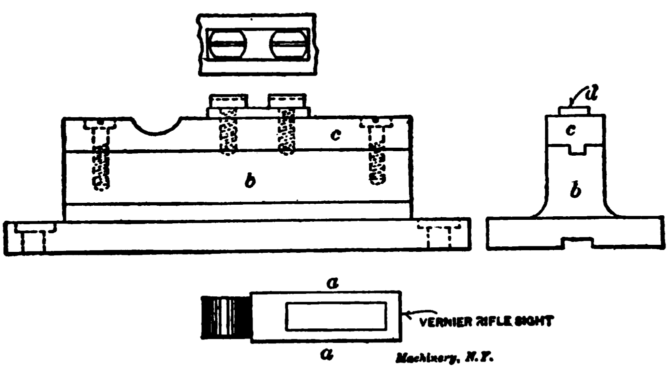

Fig. 11. Fixture for Milling Rifle Sight

tion, *c*. This is attached to the base by means of screws, after which the projection *d*, for the rifle sight, is milled in the machine used. This insures perfect alignment between the sides of the tongue, *d*, and the table travel, and in consequence the sides of the leaf are exactly parallel to the walls of the slot when the pieces are milled. In the

case of this particular piece of work it was found necessary to provide a somewhat complicated contrivance to hold the leaf down onto the fixture while milling, as the cut was rather heavy, compared with the strength of the sides of the leaf. But it was suggested that by reversing the cutters and running them down onto the work, rather than against it, the cutters would be made to hold the work down on the seating surface rather than to tend to raise it. All that was needed then was two screws, the heads of which screwed down onto the leaf. To release the leaf it was necessary to give the screw but a quarter turn, as the opposite sides were cut away to a width a trifle less than

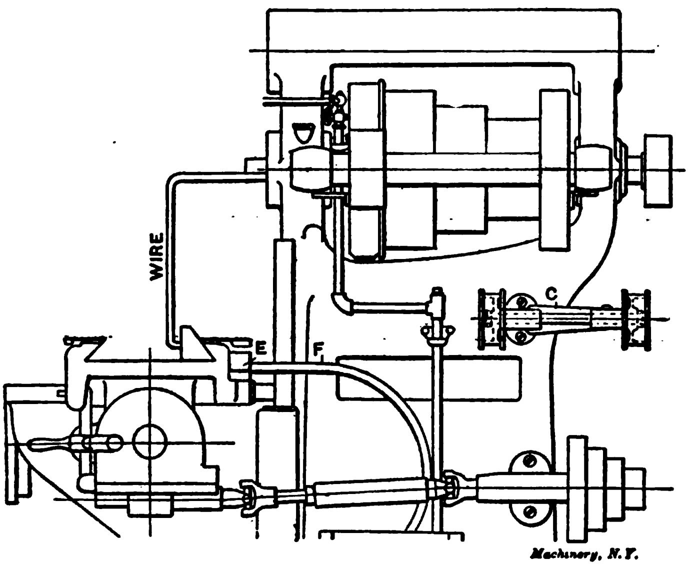

Fig. 12. Testing the Alignment of the Milling Machine Saddle

the width of the slot in the leaf. The only reason it was necessary to provide the screws was that at the ends of the cut the pressure of the cutters tended to tip the leaf.

Alignment of the Milling Machine Table

In order to produce good work when straddle-milling on a single-spindle milling machine, it is necessary to have the table travel exactly at right angles to the axis of the spindle. Should it not do so, it will be necessary to either scrape the saddle or swivel the head to get the alignment. The Lincoln type of miller usually has provision for the latter adjustment, but if not, and the saddle must be scraped, it is better to scrape the sliding surfaces which bear against the bed, instead of the table slides, unless the latter should be so badly worn as to need scraping.

The alignment of the saddle of a milling machine may be tested by means of a piece of wire attached to the spindle, as in Fig. 12.

In this case the bearing surface to be tested is on a bevel, instead of standing vertical, and therefore a cast iron block is planed to fit the angle portion, the block having a vertical surface for the point of the wire to bear against.

Principle of Gang Fixtures

Fixtures are many times made to hold two or more pieces of work to be machined at the same time, thus increasing the efficiency of the machine. Fig. 13 represents a fixture used in milling a bolt head flat on opposite sides. The fixture is designed to do away with any inaccuracy that might result from an attempt to mill bolts whose bodies were of varying sizes. For this reason the grooves for holding the bolts are made V-shaped instead of circular. The fixture is so designed as to allow the strain incident to cutting to come against the solid part of the fixture. To insure ease of manipulation, the cam levers, used in clamping the pieces in the fixture, are located in the portion of fixture nearest the operator. Were they located on the

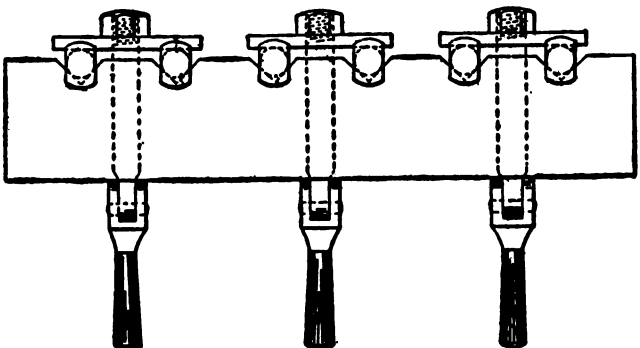

Fig. 13. Fixture for Milling Bolt Heads

opposite side it would be necessary to run the table back far enough to get the cam levers away from the cutters, so as not to endanger the operator's hands. Then again, if located nearer the cutters, they would be covered with chips, thus rendering it necessary to clean them every time before handling. The designer should always have in mind the safety of the operator, not only from a humanitarian standpoint, but also because accidents caused through poorly constructed tools and appliances are extremely costly to the manufacturing concern in whose shops they happen.

Prevention of Springing Action in Fixtures

It is generally the best practice to have the device used in binding the piece of work to the fixture connected with that part which holds the work, as shown in Fig. 14. If this plan is adopted there is no danger of springing the fixture and thus producing work which is not uniform to gage, as might happen if the design shown in Fig. 15 were used. If the fixture is extremely heavy and there is a certain amount of error allowable in the gaging, the objection to the method shown in

Fig. 15 would not be readily apparent. However, for accurate work it is advisable, when possible, to adopt the method shown in Fig. 14, for it is possible to spring fixtures which are apparently quite strong.

If a fixture is to be made in the form of an angle iron and considerable strain is to be exerted by the operation of cutting, the upright

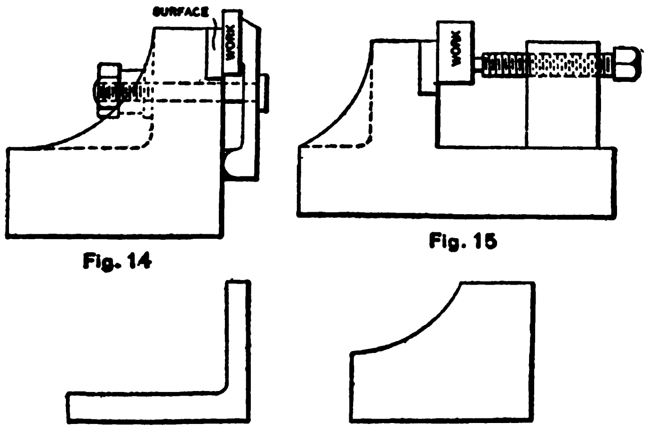

Fig. 14 Fig. 15 Fig. 16 Fig. 17

Figs. 14 to 17. Preventing Springing Action in Vises and Fixtures

portion of fixture should be made heavy, so as to absorb vibration, and it should be well braced on the back to prevent any tendency to yield under the strain. If such a fixture were made as shown in Fig. 16, the piece of work being machined would be chatter-marked from the vibration, and out of true from the yielding of the fixture. If it were made as shown in Fig. 17 neither of these troubles would be experienced,

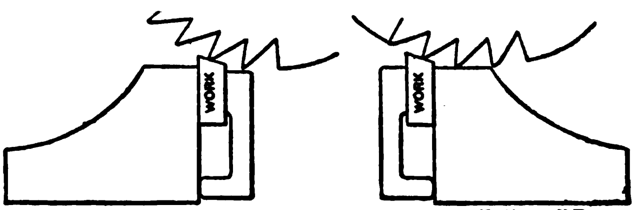

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Fig. 18 Fig. 19

Proper and Improper Direction of Cut

provided other conditions were right. When possible, the pressure of the cutter should always be against the solid part of the fixture, as shown in Fig. 18, rather than against the holding device, as in Fig. 19. One thing that is sometimes overlooked is the inability of the cutter arbor to do the work without springing. Many times cutters are made with holes so small that the arbor cannot transmit the power without springing. If arbors are made for a special job and are to be subjected to great strain, they should be as short as possible.

Fundamental Principles of Milling Fixture Design

The simplest fixture that will hold the work in a satisfactory manner is, as a rule, the most satisfactory, to say nothing of its lower cost. It is necessary at times, in order to accomplish a certain purpose, to make a complicated fixture, but the more complicated such a tool is, the greater the probability of its getting out of alignment and out of working condition. There is a tendency on the part of many young designers to make elaborate fixtures, not realizing that true success in this branch of business depends on making all machines and tools in the simplest way possible. To be sure, most of the automatic machinery on the market is very complex in design, but the designer uses every effort to simplify where possible, and still have it accomplish the desired result.

While it is absolutely necessary that milling machine fixtures be made in a manner that insures the desired degree of accuracy, yet they should be so designed that the work may be placed in and taken out in the shortest space of time possible, since this item adds very

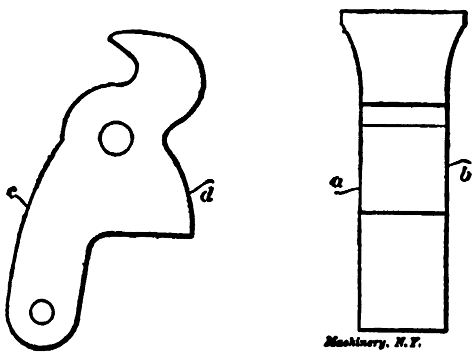

Fig. 20. Drop Forged Jaw, Finished by Milling

materially to the cost of the article. As it is customary to have the operator run several machines, the greater the length of time necessary to devote to one machine, the fewer machines he can tend.

So far as possible the design should be worked out by always working to, or by, a given surface, or other working point, and in making the fixture the same principle should be adhered to. It is poor practice to change and work from a different working surface unless compelled to do so, as any slight inaccuracy, that in itself might be of little consequence, might affect other vital portions. This same principle should apply to all machining operations.

Examples of Practical Applications

As an example of what has just been said, let us consider the cutting plier jaw shown in Fig. 20. This jaw was first forged to shape from tool steel under a drop hammer. The side marked *a* was milled first, after which the opposite side was milled. Unless great care were taken when seating in the jaws, the second side milled would not be

parallel with the first. Now, this would not materially affect the finished jaw if one particular side were selected and worked to throughout the various milling operations. The surface *a* was selected as the working surface and was the one placed against the working surface of the drill jig. Then, under normal conditions, the drilled holes would be square with the surface worked from. The same side was also placed against the stationary jaw in the milling machine vise when milling the surfaces *c* and *d*. Then, if the jaws were properly made and set in the vise and reasonable care taken to prevent the presence of chips and dirt, the surfaces *c* and *d* would be square with *a*.

A simple method to use when it is required to mill a block perfectly square is to first straddle-mill two sides by holding the block in the jaws of a milling machine vise. The other sides are straddle-milled by holding the piece in the simple fixture shown in Fig. 21, so designed

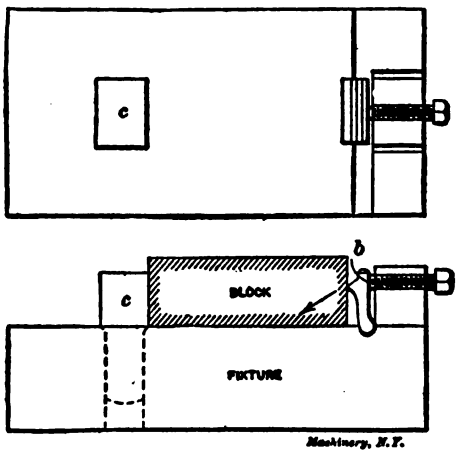

Fig. 21. Fixture for Straddle Milling

that when the piece is fastened in the fixture, the tendency of the tightening device is to draw one of the sides that were milled at the first operation down onto the seating surface of the milling fixture as shown in Fig. 21. The tilting block *b*, bearing at the bottom, acts in such a manner that when pressure is applied with the screw it forces the work down on the seating surface of the fixture, and against the upright. It might be found necessary when starting to use a fixture of this description to block up under one edge with paper to bring the milled surfaces square with the seating surface, as the spindle and table of the machine might not stand exactly parallel. This must be ascertained by experiment. The parallelism of the two may be tested with a height indicator of the description shown in Fig. 22. However, if it is found necessary to raise or lower the machine the table may not stand in exactly the same relation to the arbor as before moving. Then, again, the arbor may not be exactly true. All these things must be taken into account when testing machines for alignment.

Milling a Bicycle Hub

Fig. 23 shows a bicycle hub having projections. Through these projections, or ears, are drilled holes to receive the spokes. The equipment of milling machines in the shop where these hubs were to be milled was not sufficient to turn out the required number of pieces, and as it was not deemed wise to increase the equipment, ways were

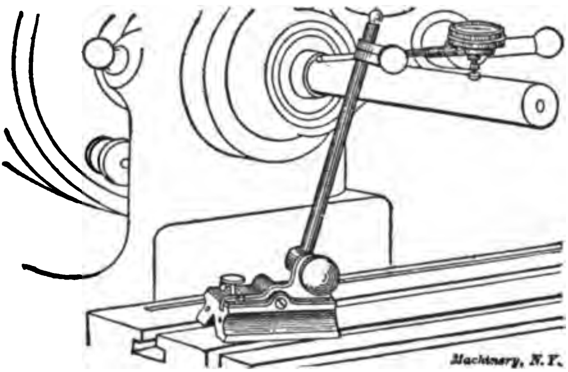

Fig. 22. Testing the Parallelism of Table and Spindle

devised of doing the additional amount of work on the machines on hand. In order to accomplish this task, it was found necessary to make multiple fixtures.

Two fixtures were made to go side by side on a plate, each fixture to hold a hub. A dog was attached to one end of the hub, the tail of the

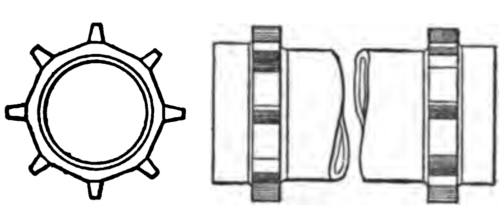

Fig. 23. Bicycle Hub

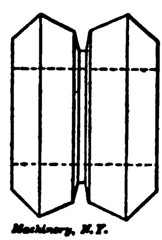

Fig. 24

dog entering an opening in the plate on the nose of the fixture spindle. On the other end of the spindle was an index plate having around its circumference a number of holes equidistantly spaced, the number of which corresponded with the number of teeth to be milled on the hub. A hardened steel pin entered these holes and thus located the hub. In making fixtures of this character where fine chips can get into the holes, it is advisable to make locating holes straight rather than tapering, since when the holes are tapering there is a strong probability of fine particles getting in the holes on one side of the pin, thus causing the work to be unevenly spaced; but where the hole and pin are straight, if the pin enters the hole, it must necessarily locate

the spindle properly. If the holes and pins are properly ground and lapped, they will retain their size for a long time. In order to facilitate the pin entering the hole the end should be chamfered somewhat.

When milling the job shown in Fig. 23, it occurred to the operator that not only could two hubs be milled at a time, but one could also make each cutter able to mill the spaces between two teeth each time, making a cutter of the form shown in Fig. 24. This shows how fixtures and methods are the results of gradual development, and almost any operation, however well planned, can almost always be still further improved upon.

Milling a Tapered Square End on an Axle or Tool

In Fig. 25 is shown a fixture used to mill a square on the end of an axle, but with the four sides on a slight taper with the axis of the

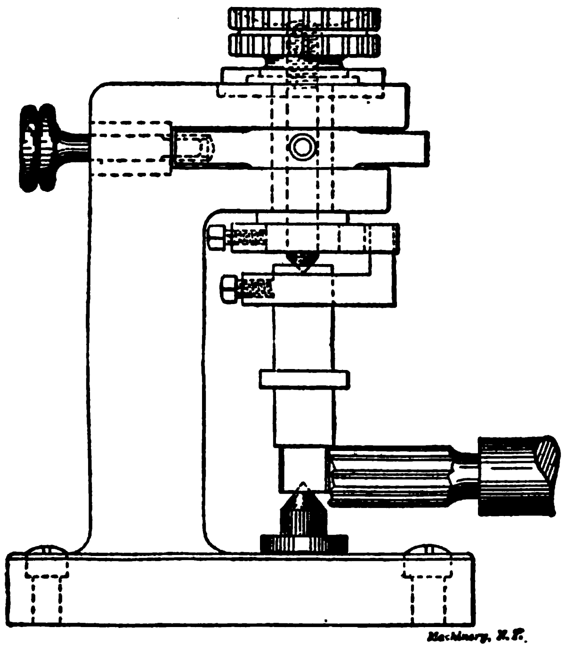

Fig. 25. Fixture for Squaring End of Axle

axle. On this account it was necessary to use an end mill rather than a face mill, and in order to use an end mill in the ordinary milling machine, the fixture must hold the axle in a vertical position and with the axle standing at the right angle to produce the proper taper. It was found to be impossible to drill the spacing holes in the indexing dial of the fixture with sufficient accuracy by holding it between the centers of the dividing head when the holes were drilled on the universal milling machine, and it was necessary to resort to another scheme. A disk about six inches in diameter was placed between the centers of the universal milling machine, and by means of an end mill

was squared. When tested with a square, it was found that the sides were not exactly square with each other, however, and they were carefully scraped until they were as square as it seemed possible to get them. The disk was then placed on a stud located on an angle plate attached to the face-plate of a lathe. The indexing dial to be drilled was then fastened to the squared disk, and after locating one side of the latter parallel with the face-plate, a hole was drilled and bored in the dial at the proper location, after which the stud was

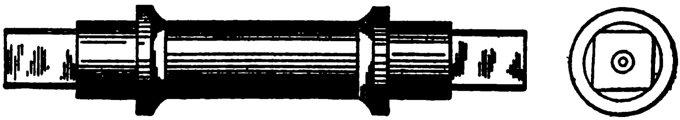

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Fig. 26. Axle Milled in Fixture Shown in Fig. 25

turned so the next side of the squared piece was parallel with the face-plate. By continuing this method, four holes were drilled and bored that were equidistant from each other. These holes were bushed with hardened steel bushings, ground inside and outside, and then forced into the holes. Pieces milled on this fixture, and which were located by this dial, were found so nearly square that no error could be detected when tested with a square. Fig. 26 represents the axle whose ends were milled.

In the previous examples an attempt has been made to avoid using complicated fixtures in illustrating the various methods of doing work, as they would be more confusing, and the simple fixtures illustrate the

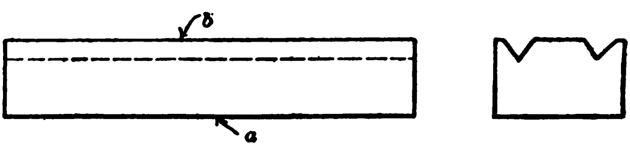


Fig. 27. Casting to be Milled in Fixture shown in Fig. 28

methods involved as well. There are certain principles which must be observed in designing fixtures of this character. These can be more plainly illustrated when simple fixtures are shown, but the designer may elaborate as much as is necessary to produce a tool adapted to the work in hand.

Fixtures with Adjustable Supports

We often have to mill articles of cast iron or other metals which are not uniform in size or shape, and which would not locate alike in any fixture, without means of compensation for the irregularities. It has been noticed that columns of milling machines, which weighed 400 or 500 pounds, have sprung out of true when on the planer table by tightening a holding bolt, when the wrench used was an ordinary 6-inch wrench, apparently applied with small force. To secure a good job, it is therefore necessary to block under the work very carefully, and then fasten it securely for the roughing cut; and for the finish cuts the strains on the clamp have to be removed entirely, or nearly so.

If it is possible to spring a large mass of metal in this manner, it is apparent that comparatively weak pieces may be distorted very easily. For this reason, it is necessary many times to provide adjustable supports at the points where the fastening devices are located, and also at points where the pressure of the cutter would have a tendency to spring the piece.

Fig. 27 represents an iron casting, the surfaces of which are to be milled. As castings will distort more or less in cooling, and as they are very liable to alter their shape when the surface "skin" is removed, it is often necessary to provide fixtures with adjustable supports for

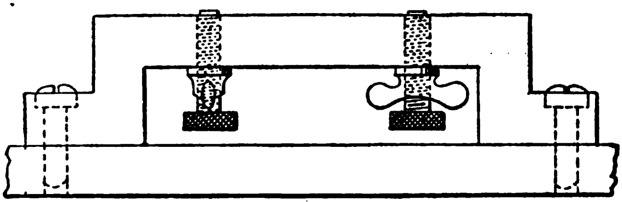


Fig. 28. Fixture for Supporting Piece Shown in Fig. 27

holding the piece, as shown in Fig. 28. In milling a piece like that of Fig. 27, such a fixture should be used when taking roughing cuts on surfaces *a* and *b*, and the finish cuts on surface *b*.

In the case of work that must be very accurate as to dimensions and truth of finished surfaces, it will be found necessary to finish the surface *a* approximately true by means of grinding or scraping before milling the surface *b* for finish. This is especially true with such work as the knee of a milling machine, as shown in Fig. 29, where it would

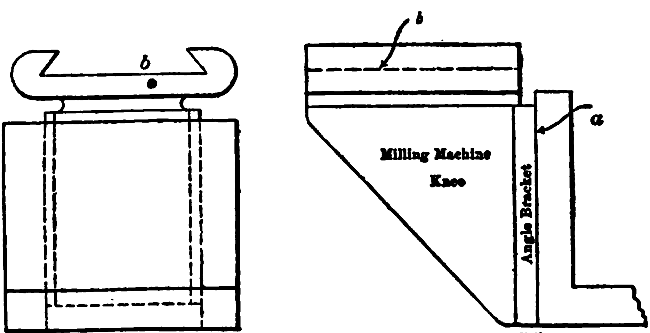


Fig. 29. Methods Used for Accurately Finishing a Milling Machine Knee

be necessary to rough mill the surfaces *a* and *b* and finish mill *a*. After this, the knee should be "rough scraped" to give it a bearing against the fixture and to prevent it winding or twisting, as would be the case if the surface *a* were not true and were clamped against the fixture. To attempt to scrape these surfaces and get out a wind occasioned by inaccurate milling, owing to one of the surfaces not being flat against the holding device, when the finishing cut was taken over the other surface, would cause much needless expense. While the

above remarks are applied directly to the milling of a milling machine knee, they are equally applicable to any piece of work that must be true, and whose shape or material renders it liable to spring as a result of some machine operation.

There are jobs which require a number of cuts on one side and which must be of a certain *uniform* depth from a given surface. If the pieces are of a uniform thickness they may be held in the usual manner, by having the under side of the piece bear against the seating

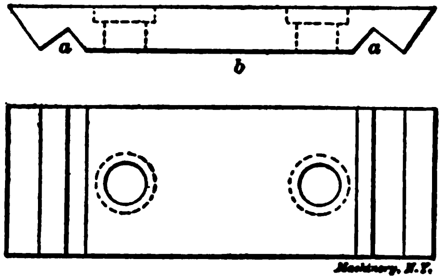


Fig. 30. Work Milled in Fixture shown in Fig. 31

surface of the fixture and the cuts taken on the upper side. If, however, the pieces are not of a uniform thickness, and the cuts must be of an exact depth, some other method of holding must be employed. Fig. 30 represents a cap used for holding a traveling carriage in place on a knitting machine. The V-groove *a a* must be given depth from the surface *b*, and owing to certain conditions it is not practicable to mill that surface at the time the grooves are milled. The distances from the screw holes must also be equal.

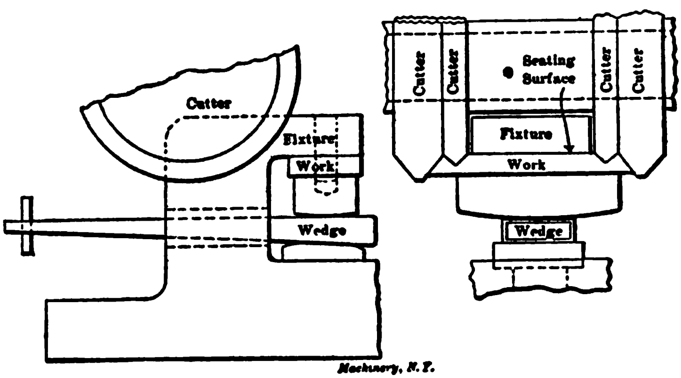


Fig. 31. Fixture for Holding the Piece Shown in Fig. 30, while Milling

A fixture of the design shown in Fig. 31 was made to hold the cap when milling the V-slots and bevel on ends. It will be observed that it is an inverted fixture and that the surface *b* of the cap, which has been previously milled, rests against an under surface of the fixture. Pins which fit the screw holes in the cap project from the seating surface of the fixture and enter these holes, thus properly locating the cap,

which is securely held against the seating surface by means of a wedge. Between the wedge and cap is placed a block, as shown. When the wedge is driven forward, the block may be removed and the cap taken from the fixture. The pin at the thin edge of the wedge prevents the wedge from being driven entirely out of the fixture.

At times when fixtures of the character mentioned are to be used, it is wise to make them of the style shown in Fig. 32, the cutters being beneath the fixture. In this case, the seating surface being uppermost, it is more easily cleaned than when the fixture shown in Fig. 31 is used.

Bridge Milling

A method of milling a certain class of work which is not used so extensively as it was a number of years ago, and which is entirely unknown to many mechanics, is known as bridge milling. In some shops work is done on profiling machines which might be done in a satisfactory manner by this method and at a fraction of the cost. The desired shape is produced by means of a form, *A*, which is securely

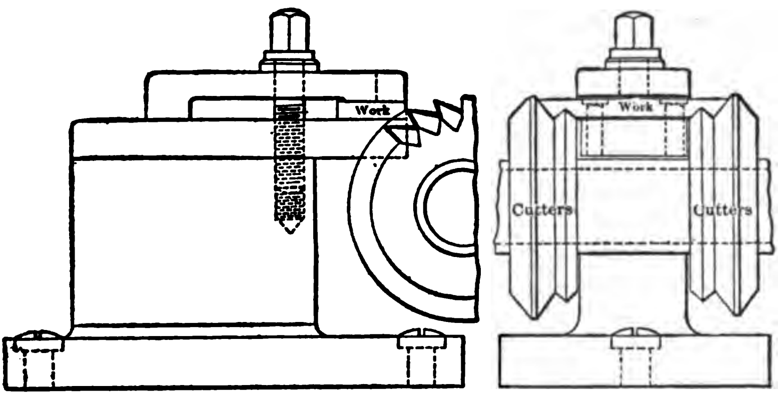


Fig. 32. Another Fixture for Holding the Work Shown in Fig. 30

fastened to the movable leaf, *B*, of the fixture, as shown in Fig. 33. This leaf is swung between two uprights, *C C*, by means of a heavy steel pin. The base of the uprights is securely fastened to the table of the milling machine by screws. To each side of the saddle, and directly opposite each other, are fastened posts, *D D*, which support the bridge, *E*, reaching across the table. The lower side of the bridge should be but a trifle above the table, say 0.001 inch, so that the table of the machine may prevent it from springing more than that amount when pressure is exerted by the operation of cutting. In the surface of the bridge is cut a slot to receive a hardened steel piece, *K K*, which, being narrow at the top, allows the movable leaf to move in conformity to the shape of form fastened to its under side.

Fixtures of this character may be used many times for milling a number of pieces at once. As an example may be mentioned a fixture for milling the legs of machinists' calipers. These are milled from pieces of square machinery steel to the shape shown in Fig. 34, where

a represents the piece of mild steel cut to length; *b*, after one side is milled to shape; and *c*, after both sides have been milled. Eight pairs of legs are milled at a time, and at a fraction of the cost of drop forgings.

Fig. 35 shows a case of bridge milling the flat portion at the end of a bicycle crank. As in the case of the caliper legs, a double fixture is used and six pairs of cranks milled at a time, milling the right-

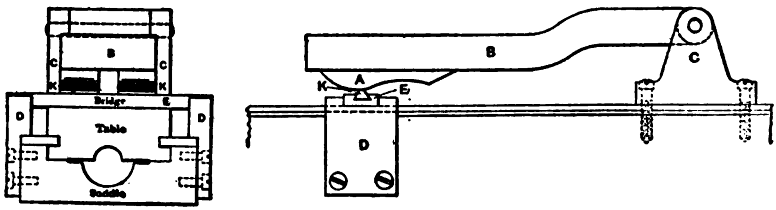


Fig. 33. Principle of Bridge Milling Fixture

hand crank in one fixture and the left-hand in the other. These are located side by side on the same machine. On account of the unequal quantity of stock removed at the various portions, a slight inaccuracy can be observed, but this is corrected by running the cutters across the work twice at the same setting of the pieces.

In these two examples of bridge milling cited, the milling was done with straight cutters, whose teeth were cut spirally, the helix being right-hand on one cutter and left-hand on the other, to do away with the thrust incidental to long interlocked spiral mills where the teeth of several cutters are of the same hand helix.

Vertical Spindle Milling

When surfaces are to be machined flat it will be found more satisfactory and quicker, in many cases, to use an end mill of the proper design. The work may be held in a special vise or in an ordinary vise

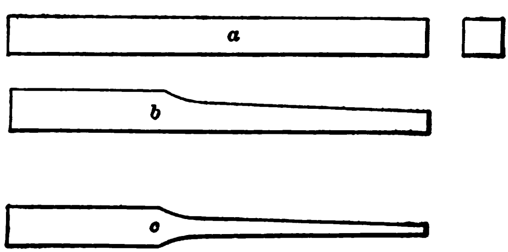


Fig. 34. Samples of Bridge Milling Cuts

attached to the vertical face of an angle iron, and done in an ordinary horizontal milling machine as indicated in Fig. 36. The best results in vertical milling are obtained by using a vertical spindle milling machine, especially if heavy cuts are to be taken; but unless there is work enough to keep the vertical machine busy, it is, generally speaking, advisable to buy a horizontal machine with a vertical attach-

ment, since it is possible to use the machine either way, as required. The fixtures for holding work when machining by this method will not differ materially from those already described. There are several advantages of vertical over horizontal milling for many classes of work; one very important one is that the surface being milled is usually more plainly in sight in the vertical machine, being turned

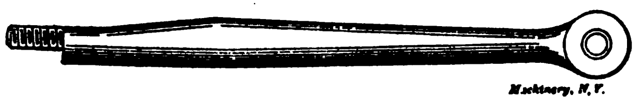


Fig. 35. Bridge Milling Out on a Bicycle Crank

upward, than in the horizontal, where it would have to be turned inward to the spindle, in order to permit the milling operation to be performed.

Cams or Eccentrics for Binding Work in Fixtures

Cams are applied to vises and special fixtures in a variety of ways and furnish a rapid means of binding the work in place. At times the cam is very simply made on the end of a piece as shown in Fig. 37. If it is necessary to get considerable length of movement to the slide of the fixture, the cam may be made on a piece having a turned projection on its lower surface, which fits in a hole in the base of the fixture. When it has been turned sufficiently to relieve the pressure against

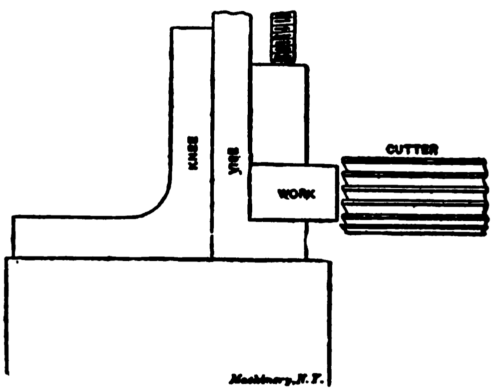


Fig. 36. End Milling on Horizontal Milling Machine

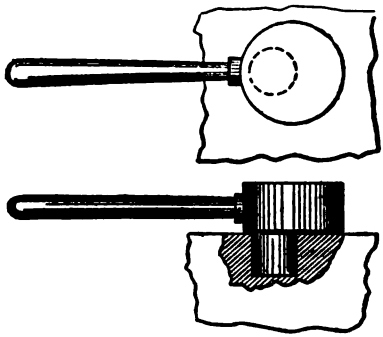
the slide, the cam may be lifted from the fixture and the slide moved as much as is necessary. After placing another piece of work in the fixture, the slide may be moved against it, the projection on the cam inserted in the hole, and the necessary pressure applied by turning the cam.

Fig. 38 shows a cam which is round in form and has a round projection which enters a hole in the fixture. This smaller projection is eccentric with the larger, in which a hole is drilled and a lever inserted as shown. This, like the previous form, may be made removable if desired. Cams of various designs may be employed for holding work, the particular design depending on the piece to be held.

Other Binding Devices

The method employed for holding work in the fixture depends, of course, on the nature of the work. Unless it is necessary to bind the work more securely than would be possible with a cam, it is not advisable to use a screw, on account of the length of time wasted in

Fig. 37. Simplest Form of Cam Binder



Machinery,N.Y.

Fig. 38. Eccentric for Binding Work in Fixtures

turning it back and forth sufficiently to secure or free the work. At times it is *necessary* to use a screw, and it is found possible to save time by the use of a collar of the description shown in Fig. 39. When the nut is turned back part of a turn, the slotted collar may be removed and the work taken out, sliding it right over the nut. After putting another piece in the fixture, the collar is placed on the screw *under* the nut, and the nut tightened to give the desired effect.

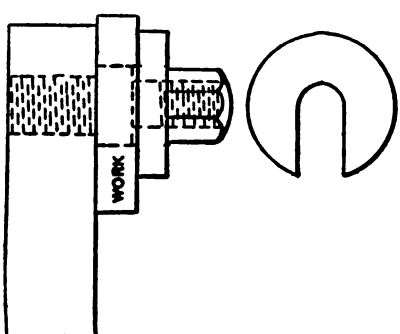
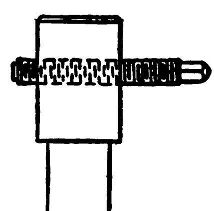


Fig. 39. Slotted Collar for Releasing Work Quickly



Machinery,N.Y.

Fig. 40. Removable Post or Stud

Fig. 41 shows a method, some modification of which may be employed to hold work when it would not do to have any screw heads or other devices projecting above the strap. When pressure is applied by means

of the screw, the portion *a* is forced down onto the piece of work. The angle piece is hinged at *b*, as shown. At times it is possible to substitute a cam for the screw, and so lessen the time necessary to operate the device. When forgings or castings are machined, it is sometimes possible to take advantage of the beveled portions occasioned by the draft necessary to get the forging out of the die, or the pattern from the mold. If the amount of bevel ordinarily given is not ample to insure desired results, a sufficient amount may be given when

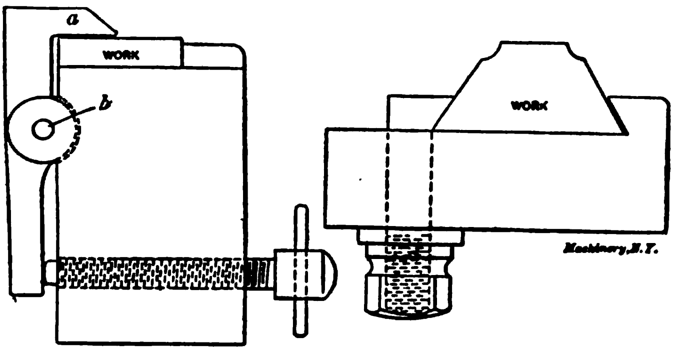


Fig. 41 Fig. 42
Holding Work where Space is Limited

the die for the forgings or the pattern is made. Fig. 42 shows a fixture holding a casting by means of considerably beveled edges.

When such a method would bind the work sufficiently strong, it is customary many times to use a screw having a right-hand thread on one end and a left-hand thread on the opposite end. Two applications

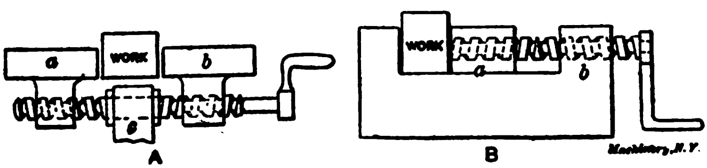


Fig. 43. Differential Screw Movements

of this principle are shown in Fig. 43; at *A* the screw is held from moving lengthwise by means of the block *c*, and the jaws are moved toward or away from each other by turning the screw. The jaw at the left, *a*, has a right-hand thread, while the right-hand jaw, *b*, has a left-hand thread. This fixture is valuable when it is desirable to mill a slot, or a projection in the center of pieces which vary in width, and where the variation is immaterial. In the fixture *B* the jaw *a* is tapped with a left-hand thread, and the stationary upright, *b*, with a right-hand thread. These threads being square in form may be of coarse pitch, thus causing the slide to move rapidly.

To save time, it is customary at times to locate the binding screw in a removable post, as shown in Fig. 40. When removing the work from the fixture the screw is turned sufficiently to relieve the pres-

sure, and the post lifted out of the hole, after which the work is removed from the fixture, the bearing surfaces cleaned, another piece put in place, and the post again put in the hole, a partial turn of the screw binding it securely. In many instances if a screw were used in a stud securely fastened to the fixture it might be necessary to give it ten or a dozen turns before the work could be removed.

Fig. 45 represents a device used for holding two pieces of work to be machined at the same time. Each piece rests against stationary por-

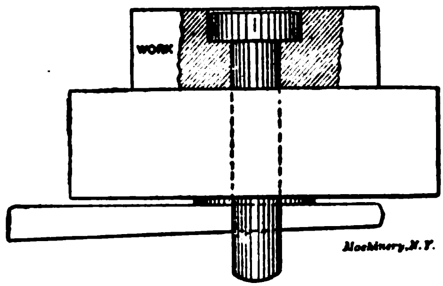


Fig. 44. Holding Work from below by a Counterbored Hole

tions *a a* of the fixture, and is held in place by the swinging pieces *b b*, which are hinged at the center, as shown, and are closed onto the work by means of the pointed screw *c* which passes through the stud *d*. This stud can turn in the hole in the fixture, and so allow the point of the screw to swing somewhat to conform to any variation in the thickness of the pieces being held. When pieces have holes through them it is possible many times to take advantage of these in holding the work. Fig. 44 represents a piece of work having on its upper por-

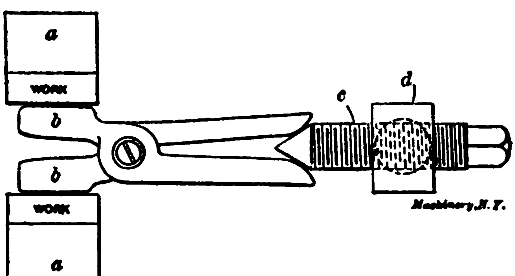


Fig. 45. Holding Two Pieces of Work at a Time

tion a counterbored hole. A pin with a head a trifle smaller than the counterbored portion of the hole extends down through the hole and through a hole in the fixture, as shown. In the small end of the pin is a rectangular hole. Through this is driven a wedge-shaped key, which draws the work solidly onto the seating surface of the fixture.

There are occasions when an ordinary cam would be objectionable and a screw would be too slow, and yet a combination of the two works nicely. Fig. 46 represents such a binding device, which is used

in holding a blank for a spring bow for a machinist's caliper, while the ends are bent in a punch press. When the screw is turned down into the threaded hole in the base, the V-shaped projection under the head passes up the incline on the upper portion of the leaf, forcing it down on the blank. When the projection of the screw reaches the flat portion at the top of the incline, the leaf has forced the blank down solidly to the bending fixture. If the screw is turned more, it, of course, continues to descend, and draws the leaf down still more. The advantage of this combination is that if a cam does not pass to its highest point at the end of the throw, it is apt to jar loose if subjected to vibration, whereas the projection under the screw head passing up

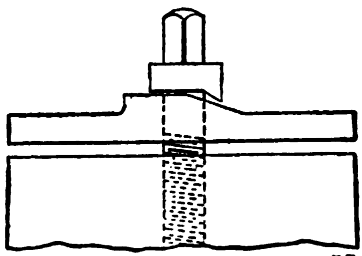


Fig. 46. Combined Cam and Screw Clamp

the incline acts as a cam, when it rests on the flat portion, and continues to draw the leaf down as the screw goes into the tapped hole. Although a fixture used on a punch press is used to illustrate the idea, the same device may, of course, be applied to fixtures for use on milling machines.

The previous paragraphs are only an outline of the fundamental principles, illustrated by means of simple fixtures and various forms of binding devices. The application must, of course, be left to the individual designer who should always bear in mind that simplicity is always preferable to elaboration, provided the simple device insures the desired result.

CHAPTER II

EXAMPLES OF MILLING FIXTURES

In the following a number of examples of milling fixture designs for definite purposes are given. These fixtures are selected as typical of the various kinds of milling fixtures found in machine shops. No attempt has been made to show only fixtures of the most approved designs, but examples indicating general practice have been taken, and attention has been called to the reasons for the special features of each design. The names of the persons who originally contributed the

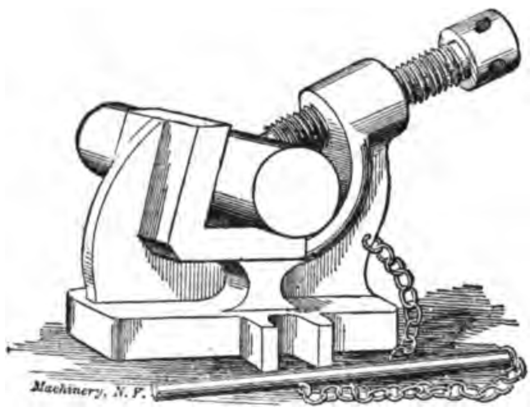


Fig. 47. Vise for Holding Shafts for Keyway Milling

descriptions of the devices shown, to the columns of MACHINERY, have been given in notes at the foot of the pages, together with the month and year when their contribution appeared.

Vise for Holding Shafts for Keyway Milling

One of the simplest designs of fixtures for the milling machine presents itself in the form of a special vise for holding short shafts and studs while milling a keyway. Such a vise is shown in Fig. 47. Several advantages over the method of clamping either in an ordinary vise or directly on the milling machine table, are apparent. The clamping bolts, holding the device to the table, are never disturbed while clamping the shafts, and if the fixture once has been set in alignment, it will remain so. Every shaft is clamped exactly alike, the screw forcing the shaft into the Vs bringing every one into exact parallelism, provided, of course, the fixture is accurately set at the start. It is obvious that this device can also be profitably used on the drill press for holding shafts and other cylindrical work for drilling, and with an adjustable arm added for holding a guide bushing for the drill, it would prove efficient as a simple adjustable drill jig.

Fixture for Holding Thin, Flat Work

It frequently occurs that thin, flat work must be held so that the whole upper surface is free, a milling cut being required to be taken across the entire piece of work. This prevents the use of any clamping devices which bear down upon the work from above, and, if the work is very thin, it does not permit of set-screws entering to bear upon it from the sides, as the diameter of the screws would be greater than the thickness of the work, and consequently project above the surface of the latter. The design of fixtures for the conditions outlined is often a rather difficult matter. A simple solution of the clamping problem is shown in Fig. 48. This cut presents merely one clamp, but it is evident that two or more clamps of the same kind are required for a complete set. The clamp is bolted to the table *H*, with the T-bolt *G*, and the bottom of the casting *A* is planed with a slot for

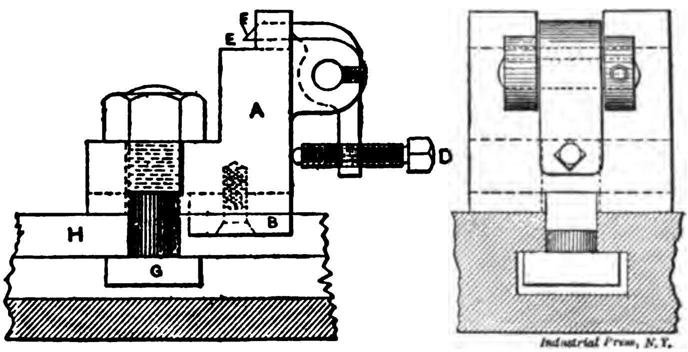


Fig. 48. Fixture for Milling Thin, Flat Work

the key *B*. These clamps are used for any flat work which is placed down on the seat *E*; the set-screw *D* is then tightened, thus forcing the steel point *F* downward and into the work. The same operation performed upon the other end securely clamps the work. If the work is long, and so thin that it is likely to spring away from the cutter if not supported in the center, this device should be modified so that the clamps are made integral parts of a fixture body which is planed on top and gives a support to the work for its full length.

Milling Fixture for Bolt Heads

In Fig. 49 is shown a device for performing a special milling operation, which, however, will illustrate some principles of general milling fixture design. It was required to mill the sides of the heads of screws, such as shown at the upper left-hand corner in the cut, so that they would assume the shape shown at the center of the cut at the top. The fixture shown in the lower part of the cut was designed for this purpose. It consisted of the body *A*, made of mild steel. It was planed all over, and a groove was cut through the body lengthwise, which was made of the same width as the diameter of the body of the screw. A hole was counterbored in the center to a depth equal to

the thickness of the head after being milled. Two clamps *B* were made of tool steel, and hardened to prevent bending. These were machined to fit the groove, thus keeping them from shifting sideways and always in line with the body of the screw to be milled. The two binding screws *C* were also made of tool steel and hardened.

Two 4-inch side or straddle mills, held apart by a collar of a width equal to the diameter of the screw, were used to mill the heads. After placing a screw in the fixture, as shown in the cut, the fixture was placed in a vise on the milling machine, the straddle mills being set to the clamps for position sidewise, and just touching the body of the fixture for the vertical position. With this fixture it was possible to mill the heads with only one cut, and it was found quite

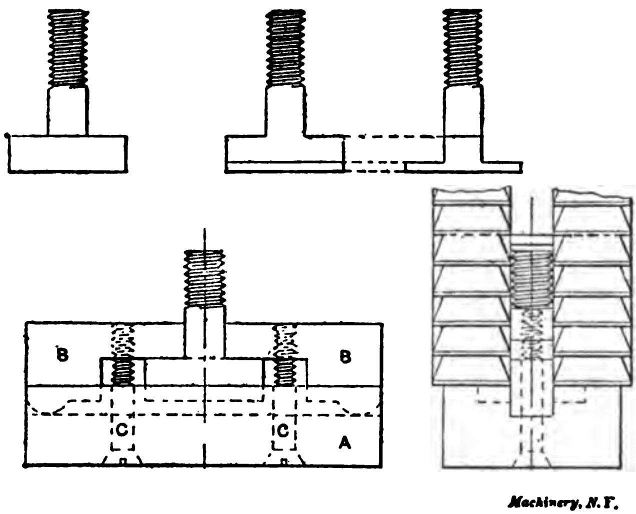


Fig. 49. Fixture for Milling Bolt Heads

satisfactory. While, however, it was possible to mill the screws in this manner so that the result was satisfactory, mechanically, it does not say that this fixture was satisfactory economically. If there were but a few screws to be milled as indicated, then, undoubtedly, a simple fixture like the one shown was preferable. But if there had been a great quantity of screws upon which this operation had to be performed, then a fixture milling one screw at a time, and requiring first the tightening of the two screws *C*, and then the tightening of the milling machine vise, would not have been in place. In such a case a fixture permitting a great number of screws to be clamped simultaneously, and to be milled all at one time, although more expensive to make at first, would in the long run have proved cheaper. A fixture employing this principle is shown in Fig. 50.

The purpose of this fixture is not the same as that of the previous device described, but the principle may be employed for almost any kind of a milling fixture for small work. The fixture shown in Fig. 50

is used on a milling machine for slotting pieces such as shown at *A*, and also for slotting screw heads. The vise jaws *G* and *H* are made out of tool steel, and are left soft; they are placed in a milling machine vise, and the piece *A* to be slotted is placed between the two jaws, as shown. The chamber *C* is a cylindrical hole into which are drilled holes from the side for the cylindrical plungers *D*. The chamber *C* is filled with tallow, and, as the pieces *A* are clamped in between the plungers *D* and the vise jaw *G*, the tallow provides an equalizing effect until all the parts are held equally firm. This means permits pieces of a slightly uneven length to be held securely. The plungers *D* must, of course, be a very good sliding fit in the holes running down to the chamber *C*. The pin *E* simply serves the purpose of locating the piece *A* by entering the hole in its center. The holes *F* are tapped to receive screws holding the jaws to the milling machine vise. Pieces *E* and *D* should be made of tool steel and hardened. When screw

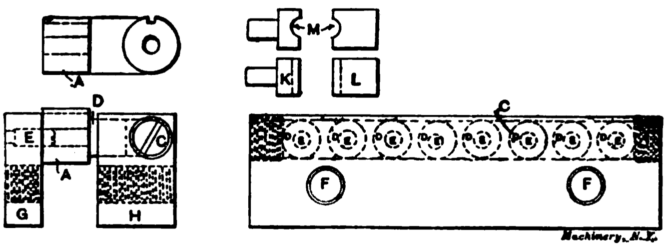


Fig. 50. Equalizing Vise Jaws

heads are slotted, the parts *K* and *L* are used instead of *D* and *E*. The screws are then held in the semi-circular grooves *M*, the operation of the device being the same as when slotting pieces *A*. The screws *I* in the ends of the circular chamber *C* simply serve the purpose of preventing the tallow from escaping at the ends.*

Fixtures for Slotting Screw Heads

While the fixture in Fig. 50 is, at times, used for slotting screw heads, it is not primarily intended for this purpose. In Fig. 51 is shown a fixture which is designed for this work exclusively, and which, although simple, is an excellent device for holding screws for slotting the heads. It has the great advantage of holding each screw with the same grip, no matter if the diameters are not uniform. It consists of the angle plates *C* and *D*, both having tongues underneath to fit the slot in the milling machine table, and in *D* are fitted the binding screw *E* and the guides *B*, which latter are securely fixed. The guides carry the V-blocks *A*, between which the screws are clamped The guides slide freely through the holes in the angle plate *C*, and may be made whatever length desired to accommodate the number of V-blocks and screws. If the full capacity of the jig is not required, say only four screws are to be slotted, as shown in the cut, the angle plate *C* is moved toward *D*, so that the binding screw *E* shall be long enough

* S. Oliver, September, 1907.

to clamp the screws. In fact, the arrangement is a most flexible one, and should prove a very satisfactory fixture for any shop.

A rather interesting and suggestive slotting device, the principle of which can be applied to a variety of work where it is necessary to slot many pieces with rapidity, is shown in Fig. 52. The part *A* is a cast iron block, which is bolted and keyed to a hand miller, and *B* is a post which swivels in *A*. At *C* is shown a lever with its fulcrum on the pin *D*. The jaw *E* is hinged to the lever and is held in a closed position by means of the spiral spring on the round head screw *F*, the tension being controlled by lock-nuts. The tool steel plate *G*, on which one

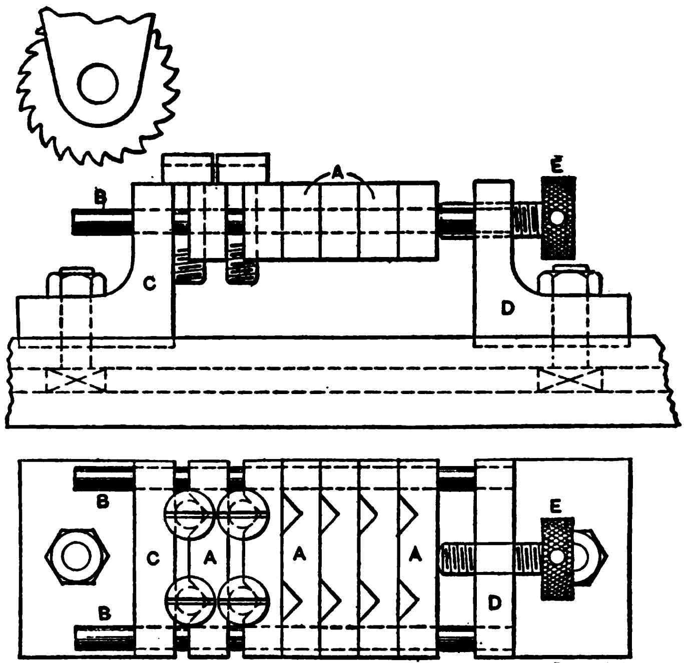


Fig. 51. Fixture for Slotting Screw Heads

end of the piece to be slotted rests, is screwed and doweled to the jaw *E*. The spring H holds the end of the lever down clear of the cutter, when it is not in operation.

The fixture is first brought clear of the cutter by moving the machine table back; the jaws are then swung out from the machine, bringing the jaw *E* against the pin *I*, which compresses the spring on *F* and thus separates the jaws, so that the piece to be slotted can be put in between the six locating pins. The pressure being then removed from the spring allows the latter to bind the piece securely in place. The lever is then swung so that the jaw *E* comes up against the pin *J*, and the lever itself rests on stop *K*. The table is then fed forward, bringing the piece against the bottom of the cutter, which slots it to the

desired depth. The piece is released by a reversal of these operations. This fixture has proved satisfactory, as it is possible when the machine is ready for operation to turn out 300 pieces per hour. The principle of this fixture could be used for slotting screws.*

In Fig. 53 is shown another device for slotting screws. This is more elaborate, and permits of a continuous operation, the operator placing the screws to be slotted in the fixture simultaneously with the slotting of the screws previously put in. At *A* and *B* are shown two rings of machine steel, case-hardened, with holes drilled on their peripheries suitable to grasp the work to be slotted. The number of holes will vary according to the speed at which the fixture is run and the work

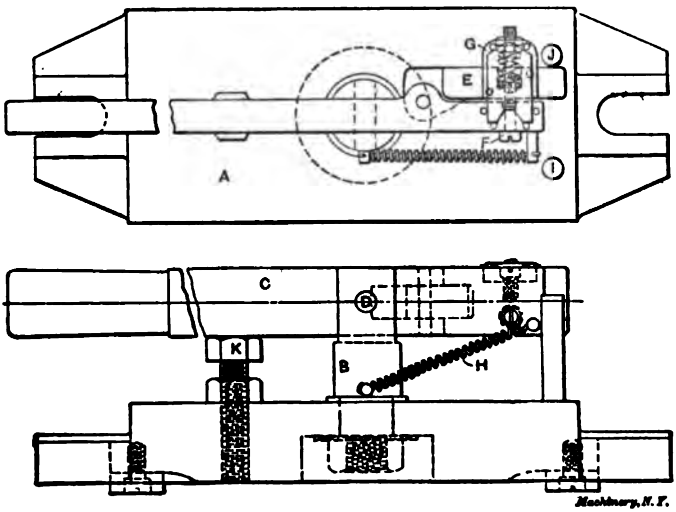


Fig. 52. Slotting Fixture for Hand Miller

being slotted. The rings are held and located on the holders *C* and *D* by screws and dowel pins not shown in the cut. Holder *D* is driven by means of a belt from the countershaft to grooved pulley *E* and through spur gears *F* and *G* and worm and worm-wheel *H* and *J*. Holder *D* is made in one piece with the worm-wheel shaft. Holder *C* is in turn driven by holder *D* by means of pins *K*, held in holder *D*. Spring *L* takes care of any variation which may exist in the size of the pieces being slotted.

To locate and drill the holes, which retain the work, in rings *A* and *B*, they are screwed and doweled on the holders, and the fixture placed on a drill press in such a manner that an equal section of the hole will be drilled in each ring. The rings are then case-hardened. For different pieces of work it is merely necessary to make different rings to suit the conditions of the piece. The bracket *M* is adjustable forward and backward to allow different thicknesses of rings to be

* S. A. McDonald, November, 1907.

used. The hole in bracket *M* is bored at an angle of 2 degrees, and the plate *A* is also faced off at the same angle, so that it will be parallel to ring *B* at *N*. By boring the hole at an angle, it will be readily seen that at point *O* the space between the two rings is the greatest, and at point *N* the least. In operating the fixture, it is placed on the milling machine so that the slotting saw will pass directly through the center of the screw head to be slotted, and directly over the center of the rings. The fixture is then started, and the operator only inserts the work in the holes. As will be seen, the piece is gripped firmly while passing under the saw, and automatically dropped when reaching the bottom.*

Fixture for Splitting Work in Two Parts

Sometimes a simple operation like splitting a piece of work in two will be found to present difficulties equal to those encountered in much more complicated operations. One such a case was met with in machining the pieces shown in Fig. 54, which were to be split in two along the line *X-X*. Owing to the peculiar shape of these pieces it was impossible to clamp them, by simple means, in any position so as to mill more than a single one at a time, and as a large quantity were to be made it was desirable to arrange so as to cut a number at a single operation. For this purpose the fixture shown in Fig. 55 was constructed and with this ten pieces could be cut at a single setting.

This fixture consisted of a casting *A* which was provided with a tongue for aligning it upon a milling machine table, and a slot at either end for receiving a clamping bolt. A series of holes, of the same size as that in the work, was drilled in the upper part of the fixture, and to insure their being parallel with the tongue, the drilling was done in place upon the milling machine, the vertical attachment being used. These holes were fitted with the studs *B*, which were of sufficient length to extend through the work *C*, as shown in the section. On the bottom of each stud was placed a split washer and nut, the latter being small enough to pass through the hole in the jig and work. These studs were prevented from turning when the nut was tightened, by means of a set-screw *D*, the point of which fitted a slot in the side of the stud. A similar slot was also cut in the side of the nut so that it could pass the set-screw. A slot *E*, the width of the splitting saw, was cut through the top of each stud, and the set-screw *D* insured this slot being always in proper position for the saw to pass through when splitting the work.

Before the pieces were placed in the jig, they were bored and faced on the top, bottom and on the straight sides, so that the splitting formed the last operation. Ten of the pieces were placed on the fixture, the bolts *B* put through and the washers put in place. Before tightening the nuts a straightedge was placed along the front side of the pieces so as to set them all squarely, after which the nuts were tightened and the saw passed through the group in the usual manner.**

* Fred R. Carstensen, September, 1907.
** Charles P. Thiel, August, 1903.

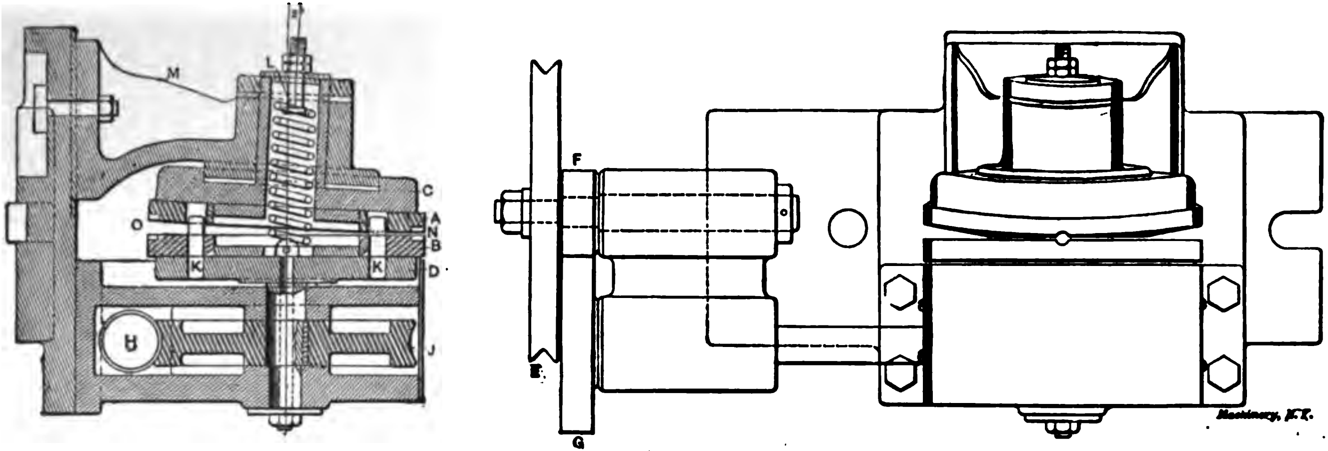


Fig. 53. Screw Slotting Fixture for Continuous Operation

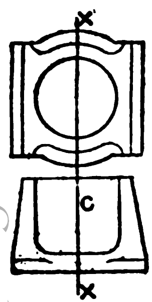


Fig. 54. Piece to be Split in Two

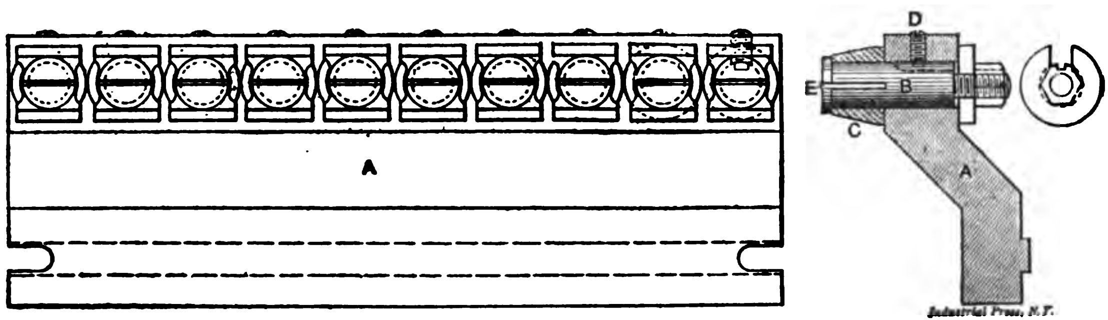


Fig. 55. Fixture for Splitting Ten Pieces at a Time

Slotting Fixture for Special Chuck

The piece shown in the upper right-hand corner of Fig. 56 is a latch chuck, made of cold-rolled steel, and used on a special machine for holding the ends of rods. The body and the center holes on both ends are turned in the lathe and the other holes are drilled in a special jig.

The fixture shown in Fig. 56 was designed for holding the chuck while milling the longitudinal slot to receive the latch, which was required to be exactly central with the axis of the piece. While not of unusual design, it possesses some advantages that make it especially useful when it is necessary to perform milling operations of this nature. It is so made as to be free from any outside incumbrances, and the parts where wear is likely to become appreciable are hard-

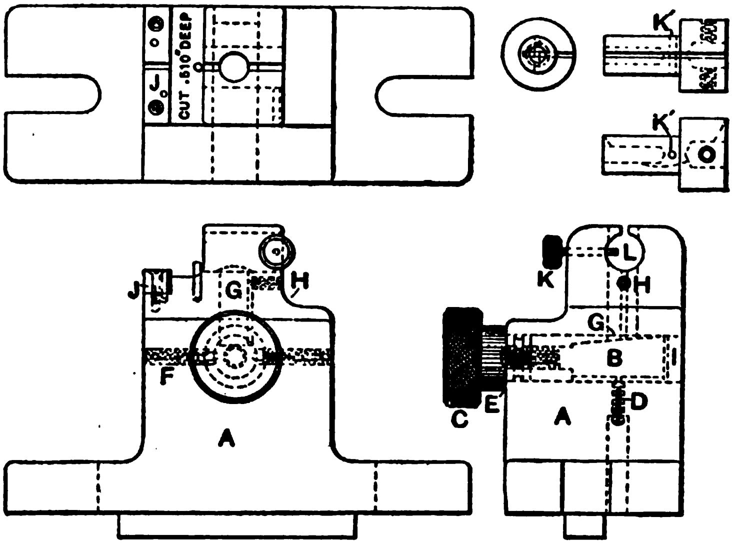


Fig. 56. Slotting Fixture for a Special Chuck

ened. The principle applied to the clamping mechanism is that of a gradual wedging action, thereby holding the work securely, and at the same time permitting the quick removal of one piece and the insertion of another by simply turning the knurled knob to the right or left as may be desired. As will be seen, the clamping mechanism is entirely enclosed, thus avoiding dust and dirt, and lessening the liability to accident from any external cause. While the illustration shows only one way in which this device may be employed, a wider field of application will without doubt suggest itself, as it is suitable for holding all kinds of milling jobs, especially where the work is polished and would be marred by clamping in a vise in the ordinary manner.

The fixture is made of liberal proportions, to insure rigidity, and is tongued and slotted for clamping bolts. Through the center of the

body of the fixture is drilled a hole carrying the tightening clamp *B*, the larger diameter of which fits snugly in the hole while the neck is turned down and threaded to fit the clamping knob *C*. To prevent the piece from turning, when the knob is turned, a groove is cut the entire length of the larger diameter, and into this fits the point of the set-screw *D*. The upper face of *B* is milled flat on a taper of one inch per foot and this part of the piece is made very hard. When clamping the work, the shoulder of the knob brings up against the body of the jig at *E*; on reversing, the two screws *F*, with their points seated in a rounded groove in the knob, prevent it from being withdrawn.

The semi-circular faced plug *G* stands vertically in the position shown, one end resting on the inclined face of *B* and the other bored out to conform to the diameter of the work. A spline is cut on one side to receive the point of the screw *H* which, while permitting a free movement up and down, checks any tendency of the piece to rotate. This plug fits the hole so freely that when it is released it falls away from the work by gravity. When the parts of the fixture are assembled, the chamber in which the slide *B* is located is filled with vaseline and the plug *I* driven in, thereby completely enclosing the mechanism and preventing the ingress of grit or the escape of the lubricant.

The milling of the slot in the work is performed with an ordinary metal saw of the required width; setting it central is simplified and facilitated by the set gage *J*, which is of tool steel, hardened and fastened in place with screws and dowel pins. The depth to be cut is measured by the graduated dial on the milling machine. In this case it is 0.510 inch, and this is stamped upon the fixture for convenience of future reference. To operate the fixture, the pin *K* is withdrawn a sufficient amount to clear the hole *L*, and the nose of the chuck to be milled is inserted against the stop pin. The chuck is rotated by hand, at the same time pressing upon the head of the pin *K*, until the pin slips into the fulcrum pin hole for the latch, *K′*, that has been previously drilled in the work. A turn of the knob *C* then clamps it tightly for the milling operation.*

Fixture for Plain Milling

While plain milling operations seem very simple to the casual observer, it is often a perplexing problem to so arrange and systematize these operations, when several surfaces are to be finished on the same piece of work, that the required accuracy is combined with a reasonable degree of speed of carrying out the work. In Fig. 57 is shown a fixture of very simple design for milling pieces in duplicate, where several faces are surfaced. This fixture reduces the setting of the machine and handling of the work to a minimum.

Let *A* represent a piece to be surfaced on spots shown on the sides and ends, these surfaces to bear definite relations to one another. It is quite possible to put spotting pieces on the top or bottom, and finish these first, fasten the work to the table and finish one side, and then, by parallels and squaring plates, finish the other surfaces from the

* C. H. Rowe, October, 1903.

first. But this means a good many measurements, bolts, straps and settings of the machine, the mass of which may be avoided by the fixture shown. It consists of a casting *B* to which the work is fastened in any convenient way after being located by the spots e e', s s and s' s', which are finished to the dimensions of the finished work, and serve to show the necessary position of the work in order to clean. The fixture has on its lower side a key slot k corresponding to the slot in the machine platen and spaced equally between the opposite spots s s and s' s' on the side.

In setting up the machine, the fixture is located by the key, and the cross-feed screw is used to bring the spots s s or s' s' to the line of cut of a face mill on the spindle nose. As the slot k is located centrally

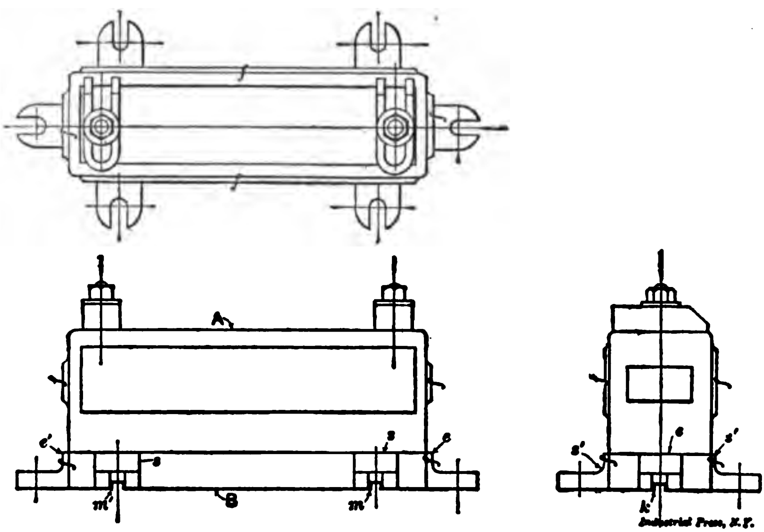


Fig. 57. Time-saving Fixture for Plain Milling

between the sides to be milled, the same setting of the machine answers for both sides, it being necessary only to turn the fixture around. The ends are placed in position in the same way, and without altering the setting of the machine, for the slots m m' near the ends of the fixture are the same distance from surfaces e e' as is slot k from surfaces s s and s' s'. Therefore, the operator has simply to see that the key enters the slot properly.

Ears may be provided for receiving the bolts which, when loosened, may simply be moved to suit the new position of the fixture as it is swung around. In practice one side may be milled first and then one end, the other side, and the other end; one rotation completing the piece. On many kinds of work the key and slots would not be accurate enough, in which case a base plate upon which the fixture might be located by dowels could be brought into service. The principle, however, would remain the same.

Simple But Efficient Milling Fixture

Figs. 58 and 59 show a piece of work, and the method employed for finishing the bosses on same by milling. In Fig. 58, which shows the work, the surfaces finished are indicated by the letter *f*. The fork end of the work is finished by the cutters *A*, *B*, and *C*, Fig. 59, while the other end is finished by the cutters *D*, *E*, *F*, and *A*, in the same figure. In Fig. 59 is also shown a plan view of the device which supports the work when being milled, as well as a side view of the device. It will be seen that the fixture for holding the work is very simple, consisting simply of three V-block supports, one at *G* supporting the casting near the fork end, and two at *H* supporting the hub at the other end. There are two vertical standards K which are mortised for a key which clamps the work down on the V-blocks. The particular feature

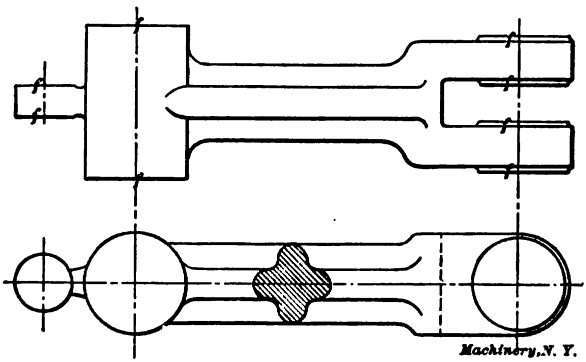


Fig. 58. The Work to be Milled in the Fixture in Fig. 59

of this device is that the V-blocks are located at such a distance from the center that, when the hub is milled and finished, and the upper plate of the jig revolved one-half of a revolution, the center cutter *A*, which has been previously employed for finishing one side of the hub, will be in correct position to mill one side of the fork end, the spacing collars between the cutters, of course, being made to take care of the required distance. A stop pin is used for keeping the upper revolving plate in the correct position in regard to the lower bed-plate.

A milling fixture of this description can be used advantageously on a great number of pieces which are ordinarily jigged two or three times. One great advantage inherent in this class of fixture is that the work is finished at one setting, thus insuring that all the machined surfaces are in proper alignment. Another advantage is that the work is handled only once at the milling machine, while if milled in the usual way, the hub end would be milled with a straddle mill, and then the casting taken to the drill press, and after the drilling operation returned to the milling machine for finishing the fork end, the work being probably held in another milling jig and located by a pin or stud through the hole in the hub.*

* Y. Ziegler, November, 1908.

Adjustable Milling Fixtures

Often, when a number of different sizes of some work, shaped and finished in the same or similar ways, are to be milled, it is possible to make milling fixtures, which with slight modifications and adjustments may serve for all the various sizes of the work, saving the expense of a great number of different fixtures. Such fixtures may be termed adjustable milling fixtures. They can often be made in a very simple manner.

The casting shown in Fig. 60 strapped to the table of a milling machine is one of a large variety of housings of widely varying shapes

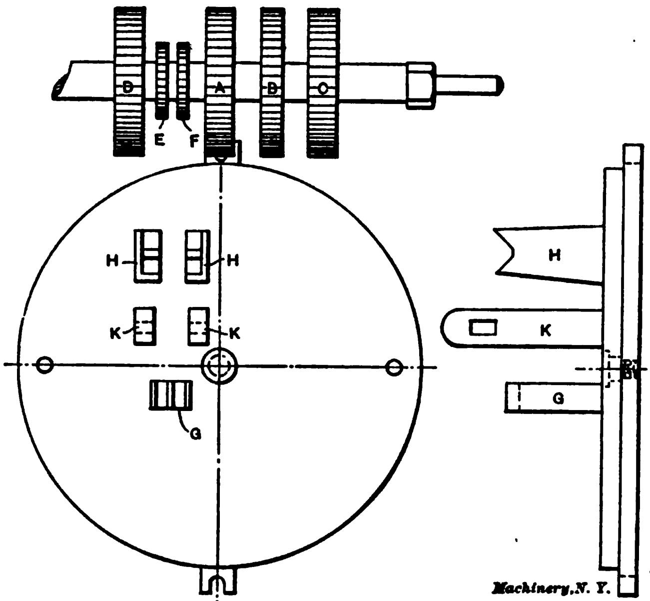


Fig. 59. Fixture for Milling Work shown in Fig. 58, at One Setting

and sizes, which are used in the construction of a certain automatic machine. These housings resemble each other in that they are provided with a V-groove at the bottom, where they are clamped to the bed of the machine, and also in the fact that they are made with various pads and bosses, similar on both sides, which have to be milled off to a uniform thickness of 1½ inch. The cross-sectioning in the plan view distinguishes the finished areas. The large number of patterns used would have made the job of providing a separate fixture for each style of casting a very costly proceeding. Therefore the following sectional fixture was made, and has proved to work well on all the different pieces on which it has been tried.

Fig. 61 shows the different parts of the fixture in detail: *A* is a block with a set-screw and spur, similar to that used on a planer; *B* is an abutment provided with a steel block to enter and hold down the V-groove edge of the casting; *C* is a simple stop to take the thrust

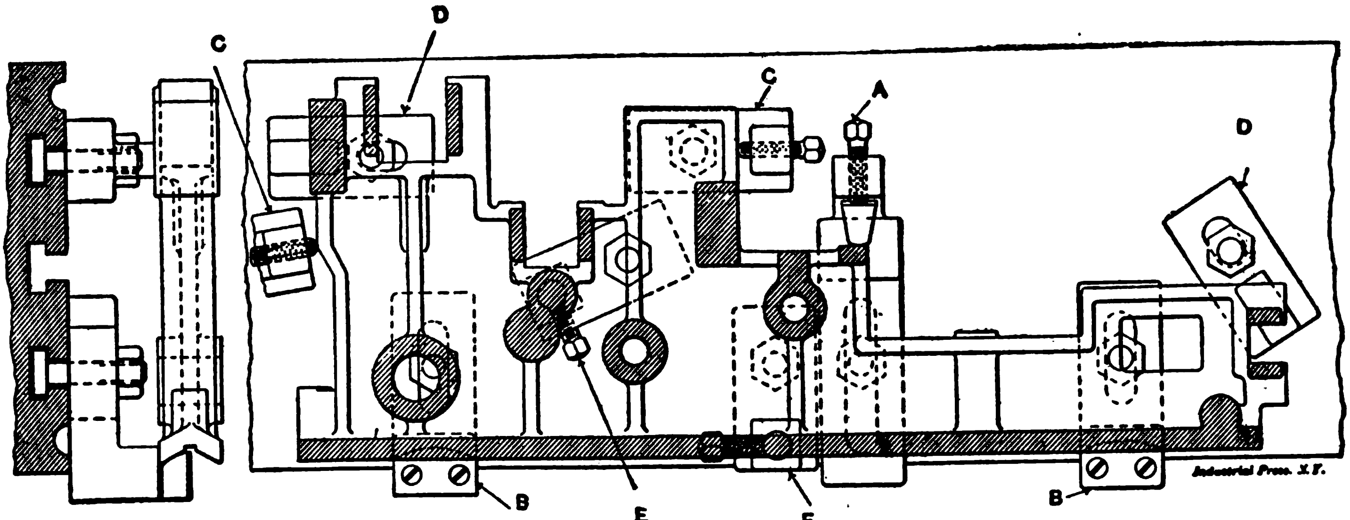


Fig. 60. Adjustable Fixture for Holding a Machine Housing on the Milling Machine Table

of the cut; *D* is a wedge used under springy places in the casting; and *E* is a spring-jack used where convenient for a similar purpose. In Fig. 60 a typical casting is shown on the milling machine platen with the various holding pieces arranged about it. The two blocks *B* are placed at the outside edge of the table, and the work is supported on these and the spur block *A*, thus giving a three-point bearing for a foundation. The spur holds it down on one side, and the steel blocks in the V-groove hold it down on the other. Blocks *C* with their set-screws are arranged as shown to take up the end thrust in each direc-

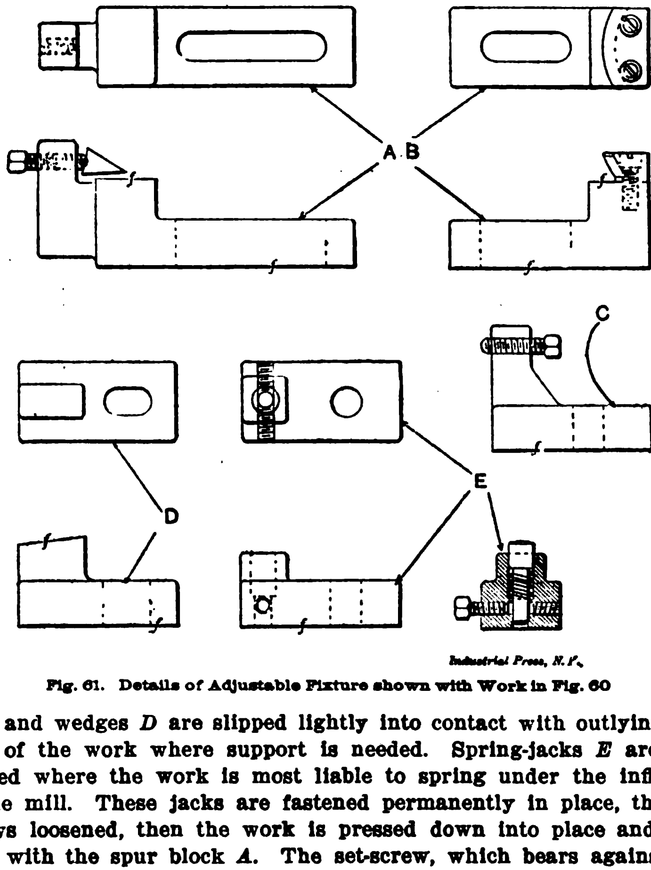


Fig. 61. Details of Adjustable Fixture shown with Work in Fig. 60

tion, and wedges *D* are slipped lightly into contact with outlying corners of the work where support is needed. Spring-jacks *E* are also located where the work is most liable to spring under the influence of the mill. These jacks are fastened permanently in place, the set-screws loosened, then the work is pressed down into place and fastened with the spur block *A*. The set-screw, which bears against the teat of the spring plug, is then clamped, and the casting is thus supported without the possibility of the casting being sprung as it would be if fastened down onto a solid bearing which might or might not be of the right height. The set-screw may be placed in either side, as convenient, as shown in detail of spring-jack in Fig. 61.

A 6-inch end-mill is used in the vertical milling attachment to make the surfacing cut. This does its work more rapidly and with less

pressure than a cylindrical cutter would. Care is taken to feed in such a direction that the thrust of the cut will be toward the V-blocks *B* or the stops *C*, although when once by mistake the cutter was run toward the spur *A*, this seemingly insecure fastening device held the work well. The housings are allowed a limit of 0.002 inch over or under the standard thickness of 1½ inch.*

Gang Milling Fixtures

In the manufacturing of small interchangeable castings for machine parts, gang milling fixtures play a very important part. When the parts are machined to extremely accurate dimensions, and are produced under the modern piece-work system, the object sought is to handle as many castings at a time as possible, in fixtures so designed as to insure the complete interchangeability of the product. To illustrate the value of gang milling fixtures for manufacturing accurately machined duplicate parts, and also how a number of such parts may

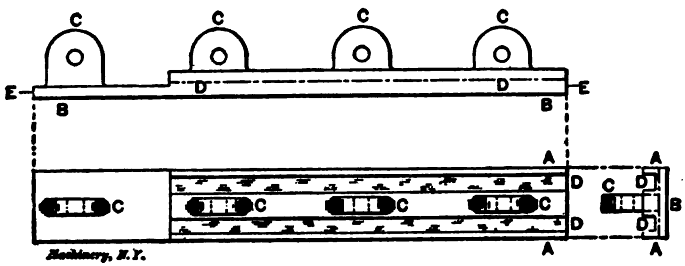


Fig. 62. Casting Milled in Fixture shown in Figs. 63 to 66

be handled and machined expeditiously at the minimum of cost, a gang milling fixture which is in use in an establishment requiring over 100,000 of the castings machined in this fixture per year, has been described in the following.

In Fig. 62 we have three views of the casting machined in the fixture. The work performed is the milling of the two channels indicated by *D*. Previous to this operation the casting is machined on the back *B* and also on the sides and ends *A* and *E* to limit gage measurements. Subsequent to the operation, the four holes are drilled in the projecting lugs *C*, the insides of the channels being utilized as banking or abutment surfaces for the locating of the castings in the drilling jig. Figs. 63 and 64 are two views of the fixture complete, Fig. 63 being the plan view, which shows the appearance of the fixture without the work in it, and Fig. 64 a vertical cross-sectional view. Fig. 65 is a longitudinal sectional view of the fixture, and also of the gang of ten cutters used in conjunction with it. *Y* represents the cutters; *X* represents the washers or collars; and *W* represents the milling machine spindle. Fig. 66 is an end view illustrating the fixture with the work in position and presented to the cutters for milling.

* Ralph E. Flanders, October, 1904.

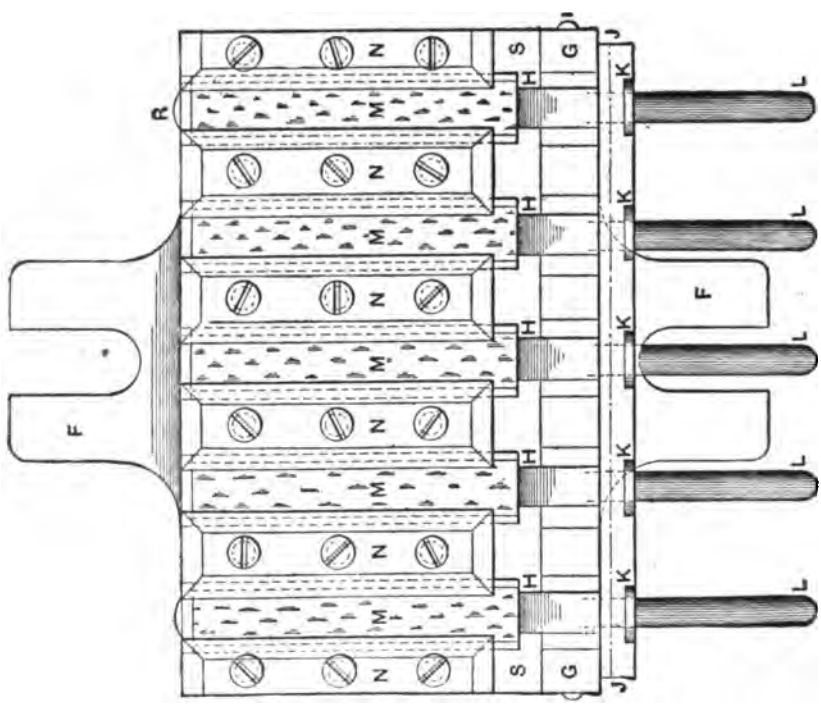


Fig. 63. Plan of Fixture for Milling Work shown in Fig. 62

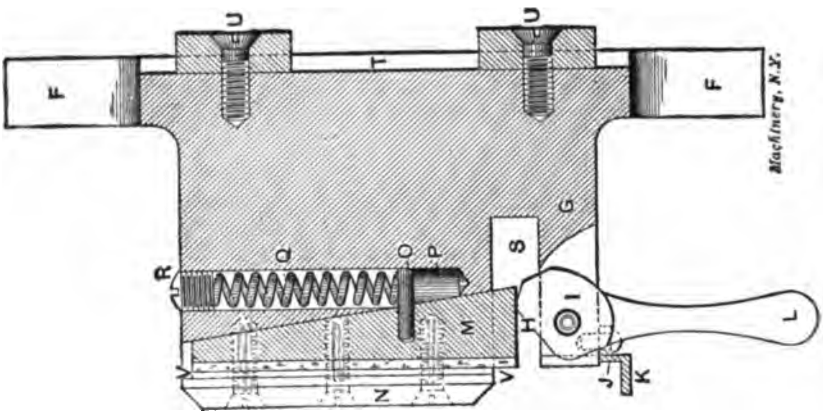


Fig. 64. Vertical Cross-section of Fixture

The fixture handles five castings at a time. The body casting has projections or wings *F*, at two sides, and has two locating tongues at *U*, for fastening and locating it on the table. The body casting has five inclined channels milled in its face to accommodate the five hardened tool steel work locators *M*. The five parts *N* are also of tool steel, hardened and tempered, and fastened to the wall surfaces between the inclined channels by means of three flat-headed screws each. These pieces serve as banking pieces or surfaces for the work to clamp up against. Five eccentric levers *L* force the work locators up the inclined ways, thus clamping the work in position against the plates

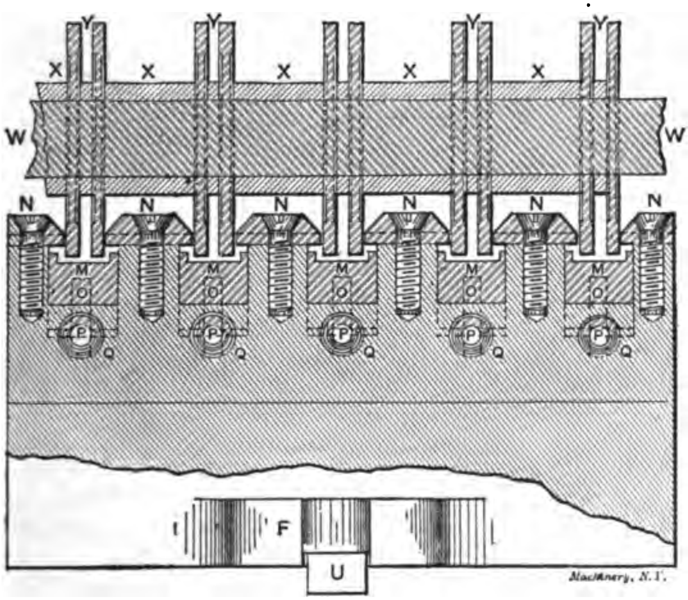


Fig. 65. Longitudinal Section of Fixture, Fig. 63, and Gang of Cutters

N. These levers are fastened in milled slots by means of the drill-rod shaft *I*. The eccentric portions of the levers are indicated clearly at *H* in Figs. 64 and 66. *J* is a stop bracket fastened to the back of the fixture or body casting by means of several round head screws. The portions at *K* are stops against which the ends of the castings to be machined, abut. The construction for forcing the work holders back in the inclined channels upon the releasing of the eccentric clamping levers *L*, thus allowing of the removal of the work, is shown in the vertical sectional view, Fig. 64. It consists of a stiff spiral spring *Q*, located in the drilled hole *P*, a pin *O* for engaging this spring, and the headless set-screw *R*. One end of the spiral spring rests against the screw *R*, and the other against the pin *O*. The tension is kept sufficiently stiff to cause the work-holders to release the work immediately upon the lever *L* being pulled upward; each of the five work holders is equipped with such an arrangement.

When in use, the fixture is clamped to the table of a large universal miller, and this is then adjusted until the work receivers or holders are in the relative positions to the cutters illustrated in Figs. 65 and 66. The castings are located in the holders; the eccentric levers are pushed downward, as shown in Fig. 66; and the castings are thus clamped in position. The feed is then thrown in and the table and

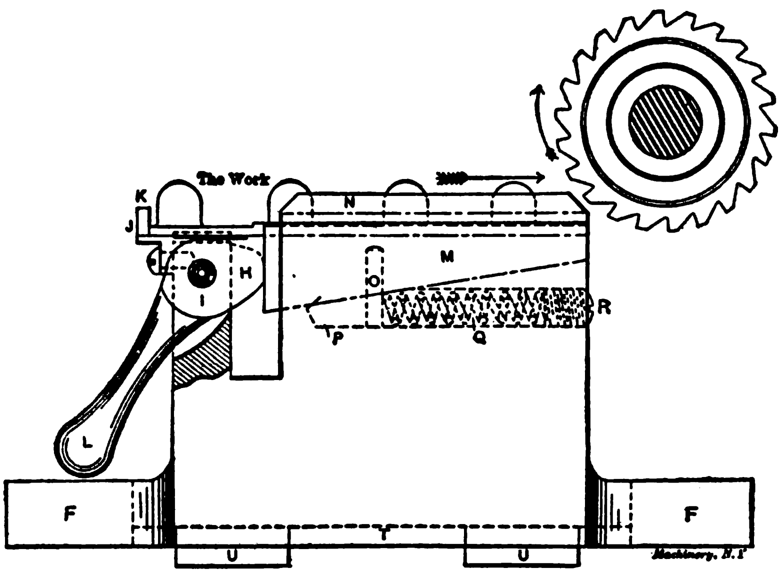


Fig. 66. End View of Fixture, Fig. 63, with Work in Position

fixture travel forward until the channels *D* are milled. The table is then fed backward and the machined work removed.*

Fixtures for Milling a Journal Cap and Base Plate

Simplicity in jig and fixture design is one of the most important fundamental principles. It is not necessary that a fixture be elaborate to be efficient. On the contrary, it is often the case that the simpler fixture is by far the one to prefer, as it has less parts to repair, and, when repairs are needed, they can be carried out with less trouble. The following description of tools used in the milling machine for finishing a journal cap and base casting, gives a few instructive examples of simplicity in fixture design coupled with efficiency.

Taking the cap first, we may hold it in the manner indicated in Fig. 67. If we are manufacturing a large number of these pieces it will pay to make special fixtures for arranging them so that the extreme length of the table feed or travel may be used. We may arrange to take one or more rows of the castings side by side, depending on the size of the miller. The cap will be seen to be resting on pins where the bosses for the cap bolts come, this making a convenient and reli-

* Joseph V. Woodworth, July, 1905.

able foundation. The cap is held sideways by the set-screws on either side and is held down on the pins by the clamp shown in the sectional view. The cut explains itself, so that but few words are necessary in connection therewith. In the holding of work on the milling machine table or in supplementary fixtures it seems to have become the idea that it is necessary to bolt it down with all the force that it is possible to use without stripping the thread on the bolts. So much strain is not necessary, serving as it does only to distort the table, making it run hard and eventually producing a permanent set which gives the working surface an untrue face. This straining of the binder bolts also wedges the T-slots out of shape, peening the metal above the T so as to project above the rest of the surface. An examination

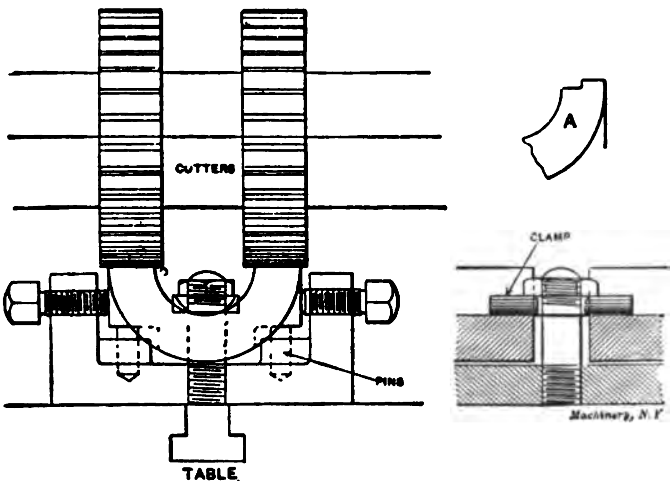


Fig. 67. Simple Fixture for Milling Cap for Bearing

of the machine in operation will show that in 90 per cent of the work done the force or pressure of the cut is symmetrical and has but little effect on the work, all the holding required being merely that necessary to keep it from sliding either along in front of the cutter or sideways. This is accomplished by bunters and toe clamps. Of course it is necessary that the work be held down on the table, but very little power is necessary in doing so. If the cap is made with the matched fit shown at *A* instead of with straight fit, the advantage of milling over planing such classes of work is very apparent, as gang cutters will then finish the work at one setting, while the planer will require at least two settings. But the real gain would be in obtaining interchangeable work which can be obtained on the planer only at the expense of considerable time and trouble, but which is a matter of course on the miller.

In performing the corresponding operation on the base casting we have the advantage of the broad base and the projecting surfaces for clamping which make it an easy matter to set and hold the work. The

same that has been said regarding the operations on the cap may be applied to the base. Fig. 68 shows how this piece would be held. The clamps hold down the piece, while the piece is blocked up against a liner to insure a setting parallel with the travel of the table. The row is kept from shifting endwise by using the bunters mentioned above.

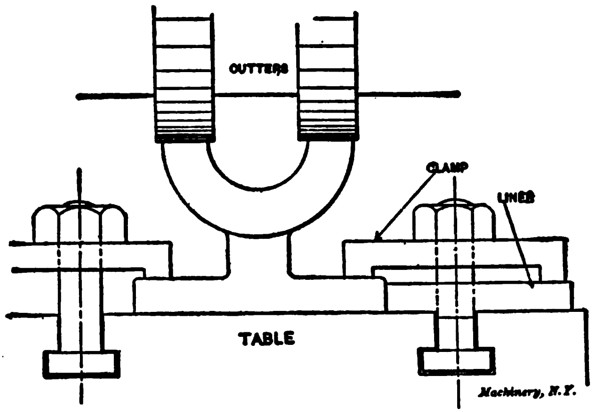


Fig. 68. Holding the Base of Bearing while Milling Seat for Cap

In machining the foot of the base piece we are confronted by a job that presents a kind of milling operation which has many little points of interest. The problem of milling comparatively broad surfaces is presented. It is an acknowledged fact that the milling of such surfaces

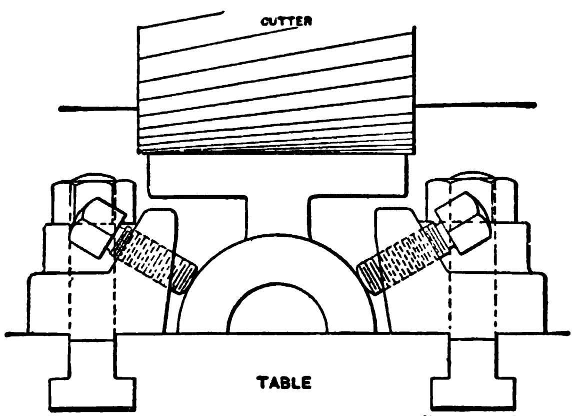


Fig. 69. Simple Holding Device used when Slab Milling the Lower Surface of the Base

must be accomplished by cutters that are so constructed that the chip is broken up into short cuts, giving the operation the advantage of the single pointed tool in the question of power required, and truth of surface obtained. This is accomplished by notching the teeth of the cutter so that they may be presented to the work successively, both

notches and teeth being cut spiral at right angles with each other. A surface produced by such a cutter will bear the strictest examinations as to truth.

Fig. 69 shows one method of machining the bottom surface. In this method we use a plain milling cutter as shown, taking one or two cuts as the case may require. If very little stock has to be removed but one cut ought to be sufficient, as the resulting surface will be good enough for the intended purpose. As will be seen the piece is held down and prevented from moving sideways by the screws which are tapped through the strips bolted to the table. This makes a convenient method and one that will be found to answer the purpose very well. Another method of performing the operation is by the use of

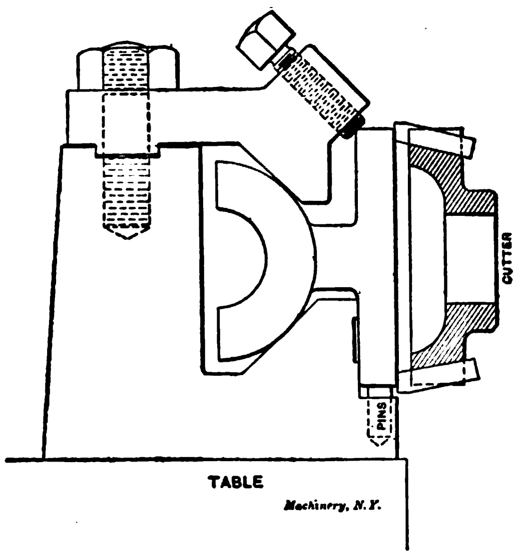


Fig. 70. Alternative Method of Holding the Base when Milling the Lower Surface

an end mill as shown in Fig. 70. This means of removing the metal is very efficient, as a very true surface can be obtained with a much faster feed and deeper cut than can be done by slab milling. The power necessary to revolve the cutter and force the feed is also very much less than that used for slab milling. While the surface may be badly marked it will yet be almost absolutely true. When the work is set up on the edge as shown, no trouble is encountered with the chips, as is otherwise the case. We are fortunate in finding this piece to be a very easy one to provide jigs for, as it permits itself to be set in almost any position. The method used in Fig. 70 is a good one, and will be found very convenient. The top clamp is removed when the work has to be removed or placed in position. This clamp serves the double purpose of holding the work and of setting it in line, the screw being used to make any allowance for variations in the castings. When this

method is chosen the machining of the bottom should be done before the cap bearing is milled, as this gives a good solid setting for the latter operation. A great many operations may be accomplished by this latter method, which are now milled with plain cutters. The action of the cutter in this operation closely resembles that of the single pointed tool and has all the advantages that are claimed for this tool, but very few of the disadvantages, it being a multiple cutter, which means greater output.

The last four cuts shown leave considerable to the imagination, as they show but an end view of the work. This is done because the same method may be used to advantage in holding one or a dozen pieces. Elaboration of the idea does not seem necessary, since the principle is shown.*

* John Edgar, November, 1906.

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NUMBER 6

PUNCH AND DIE WORK

THIRD EDITION

CONTENTS

CHAPTER I

PRINCIPLES OF PUNCH AND DIE WORK*

Under the head of punch and die work is generally included all the various tools used in blanking pieces from commercial stock; bending stock to shape; drawing out articles from sheet stock; and all the different operations performed with punching, drawing and forming presses. The most common forms of tools to be considered are the dies used for blanking articles from sheet stock, called blanking dies.

Blanking Dies

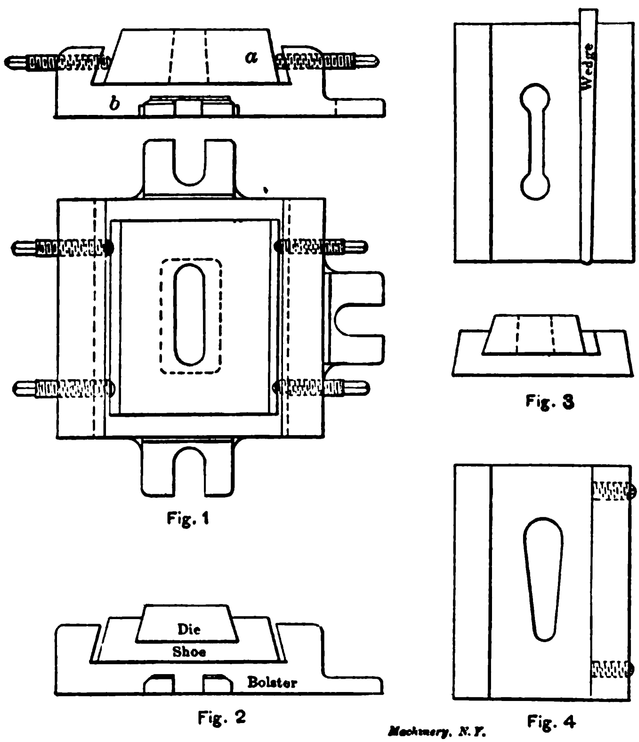
A set of blanking dies consists of a male die, or punch, as it is generally termed, and a female die, or die block. These terms are generally abbreviated and the set is called a punch and die. Blanking dies are generally considered as belonging to one of three classes: First, plain (or simple) dies; second, gang dies; and third, compound dies.

When punches and dies are used in a punch-press, and are to constitute a part of the regular equipment of the shop, they are held in suitable permanent fixtures. Dies are held in position on the bed of the press by means of a "holdfast," the name of which differs in different shops. Some of the more common names are chair, chuck, bolster, and die holder. Dies large enough to warrant it are clamped to the bed of the press, thus doing away with the necessity for holders. Dies are fastened in place in the die holder by several methods, the most common of which is by means of screws, as shown in Fig. 1, in which *a* is the die and *b* the holder. Having screws on both sides, it is an easy matter to adjust the die, loosening the screws on one side, and forcing the die over by those on the opposite side.

When the die is small, it is generally held in a shoe, as shown in Fig. 2. The manner of fastening the die in the shoe usually depends on the designer. In some shops the shoe is dove-tailed as shown, the angle being from 10 degrees to 15 degrees less than a right angle; the slot is made somewhat tapering. The die is given a corresponding taper and angle on its sides, and, to fasten it in position, it is driven securely in place. The amount of taper given the slot in the shoe must not be great, or the die will jar loose when in use. A taper of one-half inch per foot of length answers nicely. In other shops the shoe is made with a groove, as described above, only it is from 1/4 to 3/8 inch wider than the dies, which are held in place by means of a taper key or wedge, as shown in Fig. 3. When making this form it is necessary to make the dies of equal width on their ends. This method does not require so great a degree of accuracy when machining the die block.

* MACHINERY, July, August and September, 1906.

A third method consists in making a shoe having the back of the slot planed at the angle mentioned, while the front wall is made square with the bottom, the die being held with setscrews, as shown in Fig. 4. If this form is used, care must be exercised when laying out the screw holes, so that they do not come in line with the screws in the bolster when the shoe is in its proper place; and, again, the screws must not press on any portion of the die immediately in line with the opening, or it will be closed somewhat when pressure is applied to the screws. Fig. 4 shows the screws pressing on the solid portion of the die.



Figs. 1 to 4. Various Methods of Holding Work

Dies which are fastened in bolsters without using a shoe must have their sides machined at an angle, as in Fig. 1, to prevent them lifting from the strain incident to removing the punch when it has pierced the stock. The angle should be from 10 degrees to 15 degrees, some mechanics claiming best results with 20 degrees. The latter, however, seems greater than there is any necessity for on ordinary work.

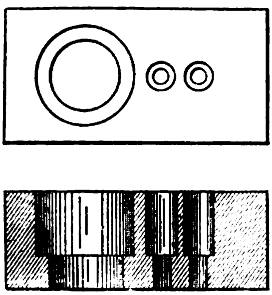
Kind of Steel Used for Die Work

For most work the stock used in making punches and dies should be a good quality of tool steel. A die that has cost from 5 dollars to 100 dollars for labor is as liable to crack when hardening as though

the same steel had been made into any other form of tool; and in fact its shape and irregular thickness of stock at various points, together with numerous sharp corners that are liable to be present, make a tool that requires extreme care in handling when hardening. A good grade of tool steel, free from harmful impurities, is less liable to crack then an inferior grade, and the slight difference in cost is offset many times by the cost of labor in the die construction. This does not necessarily mean that a *high-priced* steel must be used for this class of work; simply a *good* quality of steel, low in percentage of those impurities which cause trouble when the steel is hardened. When we speak of good, reliable steels, we do not necessarily mean high-priced steel.

If best results are desired when hardening, the steel should be annealed after the outer surface of the piece has been removed and the opening blocked out somewhere near to shape.

In all shop operations true economy should always be practiced, and many times this may be done by a saving of tool steel. If a die is



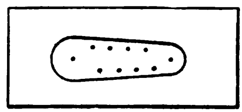


Fig. 6

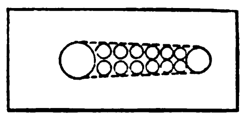


Fig. 7

Machinery, N.Y.

Fig. 5. Cast Iron Body Die, used with Tool Steel Bushings

Figs. 6 and 7. Method of Removing the Stock in a Solid Die

like Fig. 5, a saving may be effected by making the body of cast iron and inserting bushings of tool steel; and if we wish at any time to make a new die, we simply make the bushings, and if ordinary care is taken, the holes will be concentric and consequently the proper distance apart, so there will be no necessity of altering the location of the punches, as might be the case if a die made of a solid piece was hardened.

General Principles of Die Making

When a number of dies are to be made to fit the same holder, they may be planed to size in the bar and then cut apart by means of the cold-sawing machine. It will be necessary to plane again the side of dies that must fit a shoe of the style shown in Fig. 2, as one end must be wider than the other. This may be effected very readily by having a strip of cast iron planed to the proper taper to place the die on when planing or milling. The face of the die must be smooth in order that the outline traced on it may closely correspond to the templet. If the surface is a succession of ridges, the scriber will not closely

follow the edges of the pattern, and the figure traced will be larger than desired. After the face has been made smooth by planing, grinding or filing, the surface may be coated with blue vitriol solution, or it may be heated until it assumes a distinct straw or blue color, and the outline of the piece to be punched laid out.

If the die is what is known as a solid die, that is, made from one piece of stock, it may be laid off and prick-punched as in Fig. 6, after which holes may be drilled, leaving the face of the die as in Fig. 7,

Fig. 8. Die Milling Machine

after which the core may be removed. When drilling for the opening, first drill any portions which are to be left circular or semi-circular in shape. These are then reamed from the opposite side with a taper reamer that will give the desired amount of clearance. When drilling to remove the core mentioned, some tool-makers use drills of sizes that break into the next hole. After drilling all way round, the core drops out of its own accord. If this method is adopted, best results follow the use of the straight-fluted drill, Fig. 9. Others drill with drills of the size of the pilot of a counterbore, and after drilling all the holes, the counterbore is run through. Of course, it is understood

that in laying off for the holes, they are located so that the counterbore breaks into the next hole. A third method consists of laying off and drilling holes so that there is a little stock between the holes after drilling, which is broken out by means of a drift driven in from each side until the cuts meet. In this way the stock is cut away and the core removed.

After taking out the core, the die may be placed in a die milling machine, or a die sinking machine, and by the use of a tapered milling cutter the stock may be removed and the desired angle of clearance given the walls of the hole. The angle of clearance necessary for best results cannot be arbitrarily stated, but varies according to the character of the work to be done with the die. In the absence of either of the milling machines mentioned, a universal or a hand miller may be used. There are various slotting devices which may be attached to universal milling machines which are used advantageously on work

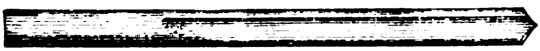


Fig. 9

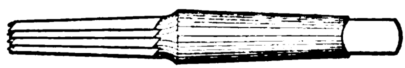


Fig. 10

Fig. 11

Figs 9, 10 and 11. Tools used in Die-making Machines

of this character. During the past few years several vertical filing machines have been placed on the market which are recommended highly for the purpose of working the openings of dies to shape. If a die milling machine, Fig. 8, is used, the form of taper milling cutter shown in Fig. 10 is employed. As the milling cutter is driven by a spindle beneath the die, the cutting portion extending up through the opening, with the face of the die uppermost, the small part of the cutting portion should be at the end of the cutter. If a die-sinking machine, Fig. 12, is used, a cutter like Fig. 11 is employed. After working the opening to shape and size as nearly as possible with the milling cutter, it may be finished by filing.

Clearance

When finishing the opening to shape and size it is necessary to get the desired clearance and to have the walls of the opening straight, as at *a a* in Fig. 13, rather than rounding as represented at *a a* in Fig. 14. The amount of clearance differs for various work and ranges from one-quarter to three degrees. The greater amount is seldom given unless it is necessary that the blank fall from the die each time one is punched. Another instance where it is desirable to give excessive clearance is where a punch with a crowning face, as in Fig. 15, is used for punching stiff stock.

When a milling machine with a slotting attachment, Fig. 17, is used, sharp corners may be cut to the line, as may certain irregular sur-

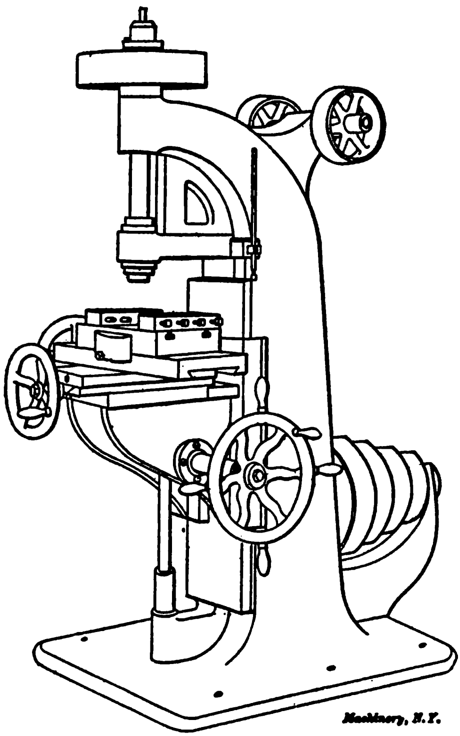


Fig. 12. Die-sinking Machine

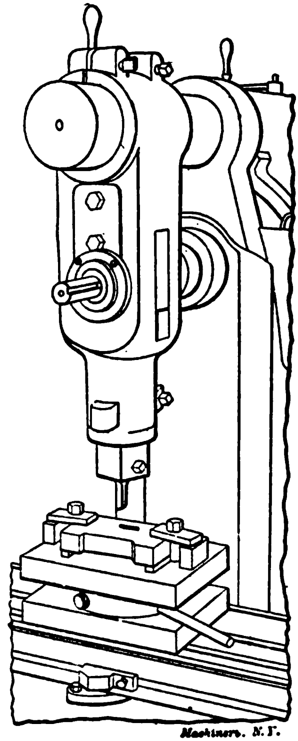


Fig. 17. Die-slotting Attachment for Milling Machine

Fig. 13

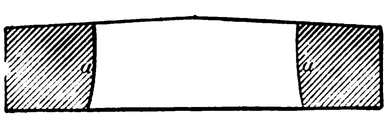


Fig. 14

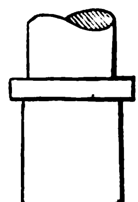
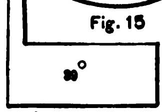


Fig. 15



Machinery, N.Y. Fig. 16

Figs. 13 and 14. Correct and Incorrect Relief. Fig. 15. Punch Crowned for Stiff Stock. Fig. 16. Templet for Gaging Relief

faces which could not be shaped with milling cutters. Of course, it would be necessary to have cutting tools of the proper shape to ma-

chine the forms mentioned, the advisability of making which would depend on whether it would be cheaper to make the necessary tools and to do the machining, or file to the desired shape. A fixture known as a die shaper, whose action resembles the slotting device described above, is made to attach to a milling machine and works the same as the other attachment.

In order to gage the angle of clearance it is advisable to have angle gages. Several of these may be made and kept in the tool chest and should be of the more common angles used. They may be of the form shown in Fig. 16, with the angle stamped on the heavier portion.

Shear of Punches and Dies

The cutting faces of dies are given shear for the same reason that the teeth of milling machine cutters are cut helical or spiral. The shear makes it possible to cut the blank from the sheet with less expenditure of power; it also reduces the strain on the die and punch. While it is customary to shear the face of the die when possible, there are instances when it is advisable to leave the face of the die

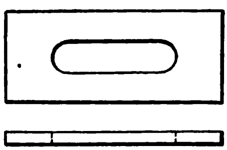


Fig. 18. A Piece of Work for which the Punch Should be Provided with Shear

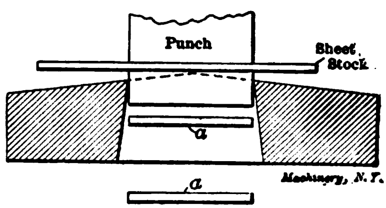


Fig. 19. A Case where the Shear should be on the Die

flat and shear the punch. The shear is given to the *punch* when the stock around the hole is the desired product and the stock removed is scrap, as in Fig. 18. The face of the die is sheared when the portion pressed through the die is the product, as at *a a* in Fig. 19, which also illustrates the shear of the die.

The amount of shear necessary to give a die to obtain best results depends a great deal on the thickness of the stock to be punched, and also on the length of the piece to be removed, and on the power of the press. The shear of a die usually commences at the center and extends toward each end, as in Fig. 19, the punch being left flat on its face. When the punch descends, the cut commences at the highest point of the die, which is in the center, and continues toward each end. The portion at the center will have been removed from the stock before the cut has progressed very far toward the ends, and in this manner the cut is distributed over the length of the piece, reducing the strain on the press and tools.

The diemaker, if he works to drawings furnished him by the draftsman, makes the thickness of die and length of punch to correspond with dimensions. However, it is customary in shops where few dies

are made and no draftsman is employed, to give the diemaker or toolmaker an idea of the shape and dimensions wanted, or possibly a templet, and he is required to go ahead and "work out his own salvation." In such cases the workman must first find the dimensions of the press to be used, the distance from the bed to the ram, the length of stroke of the ram, the amount the ram may be adjusted, the thickness of the bolster, and particulars about any shoes that are to be used. These things should be carefully set down and kept where the workman may have access to them at any time. If there are several presses, each should be marked and the dimensions of each carefully recorded, according to the work of the individual machine. If this precaution is followed and the dimensions taken into consideration when machining the die and punch, there need be none of the trouble sometimes experienced, such as a die too thick or a punch too long, or the reverse, for the press in which they are to be used.

Stripping the Stock

When articles are punched from sheet stock, or in fact from any stock where the scrap is around the punch, the stock will be carried

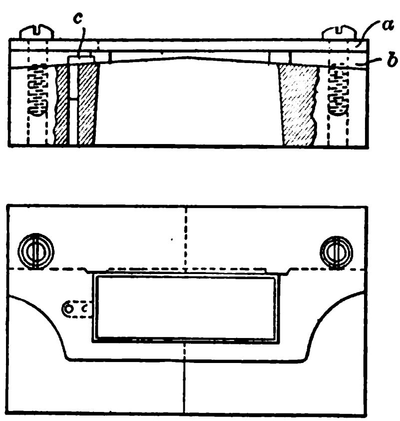


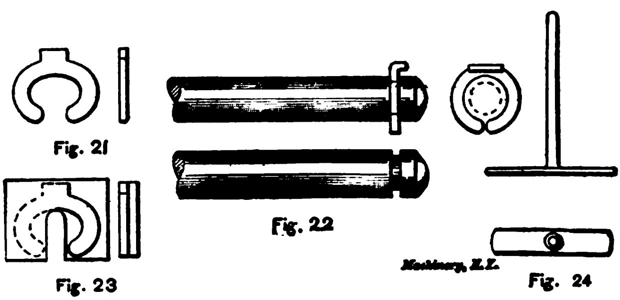
Fig. 20. Example of Stripping Plate

upward when the punch ascends, unless some device is furnished to prevent its doing so. Fig. 20 shows an arrangement *a* called a stripper, or stripping plate, the opening in this being a trifle larger than the punch. The stripper plate must be securely fastened to the die, or the die holder, and must be stiff enough to prevent its springing when in use. Between the stripper and the die (at *b*) is a guide against which the stock being operated on rests, and which determines the amount of scrap at the back edge of the sheet. This guide is made of a thickness that insures the space between the die and stripper being somewhat greater than the thickness of the stock used; in fact, the space must be sufficient to allow the stock to move along

easily even when the surface is made somewhat irregular by the operation of punching. At *c* is a guide pin or stop against which the stock is placed to determine the endwise setting.

Templets

When dies are made for producing pieces that must be of a given size and shape it is necessary to have a piece of the same shape and size to work to; this is called a templet. At times it requires a considerable degree of skill to make a templet that will answer for the work in hand. As an example, the templet shown in Fig. 21 may be referred to. After blanking and turning the ear at the top of the piece to be made, it was to be closed on a groove in an axle, as shown in Fig. 22. After closing, the outside of the washer was supposed to run about true. The die was made to a templet and it was found less difficult to make the die than the templet. In this instance it was necessary to make two pieces of the desired shape exactly alike, one of which was closed on the model axle and tested. The points that were not right were located on the one that had not been closed up. Then others were laid out from it, due allowance being made for the im-



Figs. 21 to 24. Example of Templet Making

perfections of the first. When making, two pieces of stock were placed together, and one half was worked to the laying out lines as in Fig. 23. After the other half had been blocked out somewhere near the line, the pieces were reversed and each half that had been blocked out was finished to the finished half, as indicated. In this way the ends were exactly alike and the two being machined, or filed together, were, of course, alike. When one was forced down or closed on the axle and was found correct, the other answered for the templet to be used in laying out the die, and afterward to fit the opening, too. While the example related was comparatively simple, it did not appear altogether simple to one not accustomed to that class of work, and it serves to illustrate the idea brought out.

In order that templets may be easily handled, it is customary to attach some form of handle to them, which is sometimes done by drilling and tapping a hole in the templet, and cutting a short thread on a piece of wire which is screwed into the tapped hole. Another common method is to attach a piece of wire by means of a drop of solder, as

shown in Fig. 24. This method is open to the objection that the wire must be removed from the templet when it is used in laying out the punch, as it is necessary, when the templet differs in shape on two edges, to lay opposite sides of the templet against punch and die.

Sectional Dies

Dies are many times made in two or more sections in order to facilitate the operation of working the opening to shape. In other cases the die, if solid, would be so large as to render it well-nigh impossible to harden it in a shop with only the usual facilities for doing work of this class. And then again if it should go out of shape in hardening, it would be a difficult task to remedy the defect. If made in sections, as shown in Fig. 25, it would be possible to peen or grind to the original shape with little trouble.

A die of the design shown in Fig. 26 may be made sectional because it is much easier and cheaper to make than if solid. The sec-

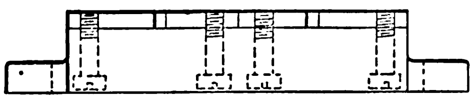


Fig. 25. Sectional Die held by Screws

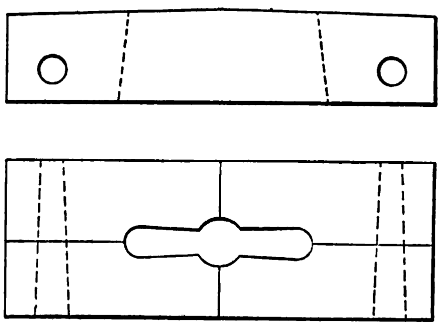


Fig. 26. Sectional Die Located by Taper Pins

tions are held in their proper location by dowel pins. They are held together by the shoe which secures them in the press. If the die is comparatively small, the circular shapes at each end and center are produced by first drilling, and then reaming from the back, with a reamer of the proper angle. The sections may be separated and the balance of the stock removed in a shaper, planer or milling machine. When this stock is removed the die may be held at the proper angle to produce the desired clearance. After machining as close as possible, the surfaces may be finished with a file and scraper.

When the opening has been finished to the templet, the top may be given the proper *shear*. In order to facilitate the operation of grinding when the die is dull, the stock may be removed, as in Fig. 27, leaving about ¼ inch on each side of the opening at the narrowest portion.

There are certain forms of dies where it is not feasible to cut away a portion of the top, as shown, but where it can be done it saves much time when grinding.

Correcting Mistakes Made in Dies

Should the workman, through misunderstanding or carelessness, make the opening too large at any point, he should not attempt to

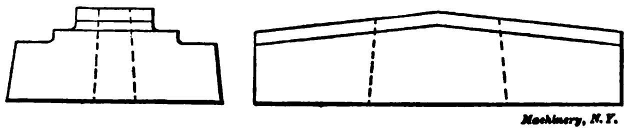


Fig. 27. Method of Cutting away the Top of Die to Facilitate Grinding

peen the stock cold, as is sometimes done, for while it is possible to do this and then finish the surfaces in such a manner that it will be scarcely noticeable, the stock directly below where the peening took place will almost surely crack during the life of the die.

Should the mistake referred to occur, heat the die to a forging heat, when the stock may be set in without injury to the steel. When set-

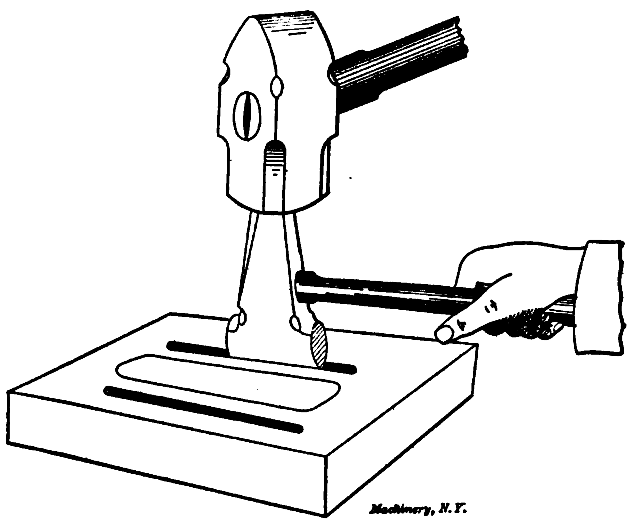


Fig. 28. Closing up a Die which is too Large

ting in, a blacksmith's fulling tool may be used, this placed on the face of the die and struck with a sledge, as in Fig. 28. If there is objection to disfiguring the top surface of the die, this method can, of course, not be used, but if the top is to be cut away, as shown in Fig. 27, the depression made by the fulling tool would be entirely cut away. It is never good practice to bend, set in, or otherwise alter the form of steel when cold, if it is to be hardened, as such attempts nearly always end in a manner entirely unsatisfactory.

Reworking Worn Dies

When a die becomes worn so that the opening is too large, or the top edge of the walls of the opening are worn so that the die is "bell muzzled," it may be heated to a forging heat, set in with a fulling tool, or a punch of the desired shape, after which it is reheated to a low red and annealed. After annealing it is reworked to size. This reworking, care and judgment being used, gives excellent results, and effects a considerable saving, as otherwise it would be necessary to make new dies, while the die may be reworked at a fraction of the expense of a new one.

When making a sectional die, it is possible in case the opening is a trifle too large, to work a little stock off the faces that come together, provided the outer edges have not been planed to fit the holder; also,

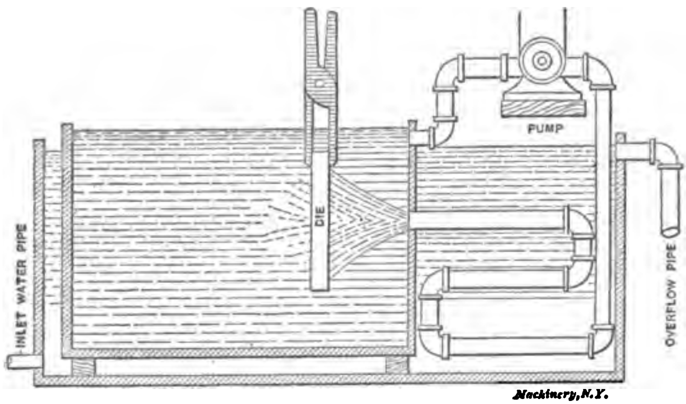


Fig. 29. Arrangement of Oil Cooling Bath

if it is allowable, these surfaces may be cut away the desired amount, and a strip of stock of the proper thickness placed between the die and holder. Considering the liability of a mistake taking place when the beginner is doing work of this kind, it is, generally speaking, advisable to leave the fitting of the die to the holder until the opening has been worked to size.

Hardening Dies

There is probably no one article the hardener is called on to harden that he dreads any more than a die. If he succeeds in bringing it out of the bath without a crack, he gives the credit to "luck"; and if it cracks, it is almost what he was looking for. This is an unfortunate condition, as there is no need of losing dies in the operation of hardening. Of course, if a piece of imperfect steel is used, it is almost sure to go to pieces in the bath; but if the steel is of the proper quality and in good condition, there need be no trouble when hardening.

When handling work so diversified in character as the class under consideration, the operator should not assume that it is possible to

adopt any set method which is not to be deviated from, as there is no one class of work that calls for a greater exercise of skill and common sense than the proper hardening of punch-press dies, unless it be the hardening of drop-forging dies. For most dies of this character, however, and especially for those complicated in form, and which must retain as nearly as possible exact measurements, there is no method that will give the satisfaction derived from the method known as "pack-hardening."

Pack-hardening

When pack-hardening such pieces, best results are derived from the use of a bath of raw linseed oil of the type shown in Fig. 29, in which the oil is kept from heating by being pumped through a coil of

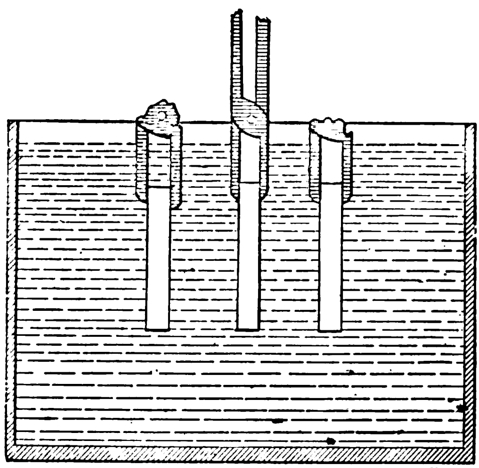


Fig. 30. Dipping the Work in the Bath

pipe in a tank of water, and then forced into the bath and through the opening as shown. If such a bath is not at hand, good results can be obtained where the oil is not agitated but the die is swung back and forth and moved up and down somewhat in the oil. If many dies are to be hardened this way, however, it is necessary to have a bath of generous proportions, or else several smaller baths, as it would not do to use the oil after it becomes hot, although oil that is heated somewhat will conduct the heat from steel more rapidly than would be supposed, and is better adapted for hardening than if it is extremely cold.

General Directions for Hardening

The secret of success in hardening dies by the ordinary method consists in getting as nearly as possible a *uniform* heat. To accomplish this the die cannot be heated very rapidly, as the edges and lighter portions would heat more rapidly than the balance of the piece. Unequal contraction, when quenching in the bath, follows un-

even heating, and unequal contraction causes the die to crack. High heats cause cracks in steel. Then, again, high heats render the steel weak, and as a consequence it cannot stand the strain incident to contraction of one portion of the steel when another portion is hard, and consequently rigid and unyielding. Steel is the strongest when hardened at the proper temperature, known as the refining heat.

Cold baths are a source of endless troubles when hardening dies. They will not make the steel any harder than one that is heated to a temperature of 60 or 70 degrees, or even warmer than this, but they will cause the die to spring or crack where the warmer bath would give excellent results. A bath of brine is to be preferred to one of water for this class of work, the brine being heated to the temperature mentioned above.

Have the bath of generous proportions. When the die is properly heated, lower it into the bath as shown in Fig. 30, moving it slowly back and forth to the positions shown, which causes the liquid to circulate through the openings, thus insuring the walls of the opening hardening in a satisfactory manner. Then again, moving back and forth brings both surfaces of the piece in contact with the liquid, causing them to harden uniformly, and preventing an undue amount of "humping," as would be the case if one side hardened more rapidly than the other. The workman must, of course, exercise common sense when doing this class of work. If he were to swing a die containing sharp corners, intricate shapes, and fine projections as rapidly in the bath as it would be safe to do were the opening round or of an oval shape, it might prove disastrous to the die, as such a shape would give off its heat very rapidly, and as a result the fine projections and sharp corners would harden much quicker than the balance of the die; and as they continued to contract, the projections would fly off, or the steel would crack in the corners. To avoid this, have the bath quite warm, move the die slowly, and as soon as the portions desired hard are in the proper condition, remove the die and plunge it in a bath of warm oil, where it may remain until cooled to the temperature of the oil.

Most of the trouble experienced when hardening dies is occasioned by one of two causes—possibly both. The first cause is uneven heating, the second, cold baths.

The Punch

The method of holding the punch depends on its shape and the style of die, as well as on the holders at hand in the shop. If it can be made as in Fig. 34, with a shank to fit a holder which enters an opening provided in the lower end of the ram, it will be comparatively simple to make. At other times it will be necessary to attach several punches to a holder, as in Fig. 31. When these punches can be attached to the holder by means of round shanks it will be found a satisfactory method. For many forms of punches, however, this would not answer, it being found necessary to attach them by screws, dowel pins being provided to keep them in position, as in Fig. 32 at *a*. Then, again, it is sometimes thought advisable to use a fixture for

holding the punches, having a dove-tailed slot cut in the face as in Fig. 33, the punches having a tongue which is fitted in the slot. The punches are securely held by means of setscrews. As the opening in the lower end of the ram to receive the punch holder of small presses is ordinarily square, the holder is made of a shape that fits the opening, the hole to receive the punch being round. At times the holder is

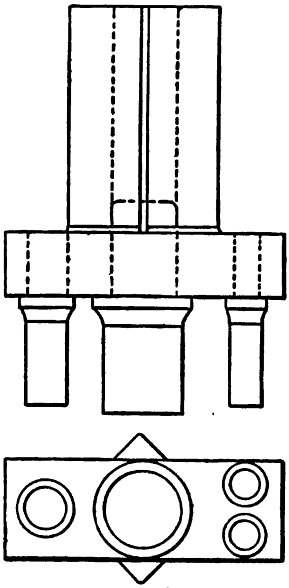


FIG. 31

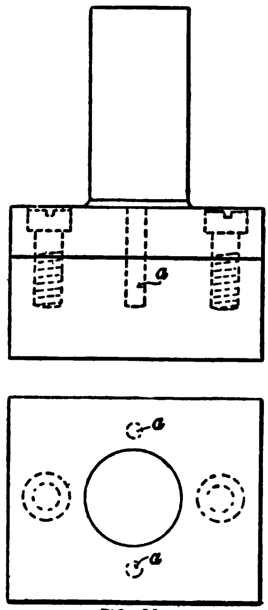


FIG. 32

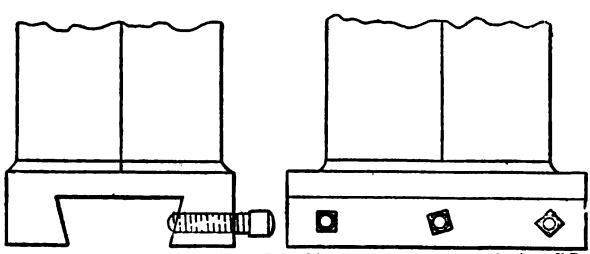


FIG. 33 *Machinery,N.Y.*

Figs. 31, 32 and 33. Various Methods of Holding Punches

split as in Fig. 35. When pressure is applied, the holder is closed onto the shank of the punch, thus holding it securely. At other times the holder is made without splitting, and a setscrew placed in the lower end of the holder, Fig. 36. This setscrew, when screwed against the punch, holds it securely in place.

It is customary to make the die, and harden it, and then make the

punch and fit it to the die. After squaring the end of the punch that is to enter the die, the surface is colored with blue vitriol solution, or by heating it until a distinct brown or blue color is visible, after which the desired shape is marked on the face by scribing. If it is considered advisable to lay out the shape by means of the templet, it may be done; but if the templet is not of the same shape on its two edges, or the ends are different from one another, it will be necessary to place the opposite side against the punch, from that placed against the die when marking. However, it is the custom many times to mark the punch from the die. If the die is given shear, it is necessary to mark the punch before the face of the die is sheared. When

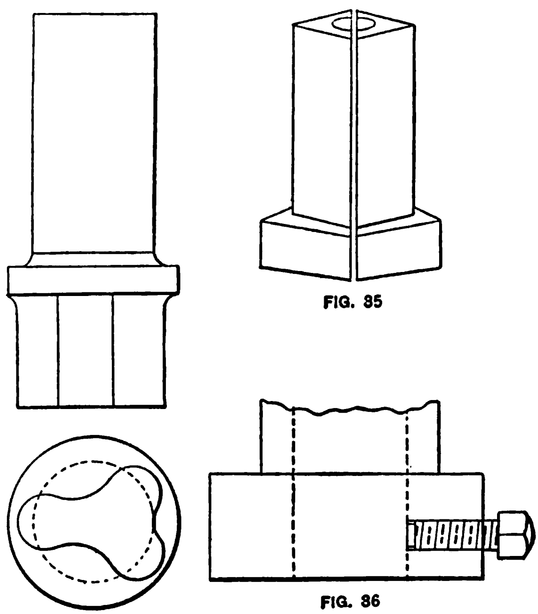


FIG. 35

FIG. 36

FIG. 34

Machinery,N.Y.

Figs. 34, 35 and 36. Punch and Punch-holders

laying out several punches from a die which has a number of impressions, it is necessary to lay out the punch from the die.

The surplus stock on the punch is removed by filing, chipping, milling or planing, as the case may be, until it is but a trifle larger than the opening in the die. The end is then chamfered somewhat so that it enters the opening, and the punch is forced into the die a little way. It is then removed, the stock cut away, and the punch forced in again, this time somewhat further. This method is continued until the punch enters the die the required distance. It is then filed or scraped until the desired fit is obtained. When punch and die are to be used for punching paper, soft metals, or thin stock, the punch must fit nicely. If the stock is thick, or stiff, the punch may be somewhat looser.

For stock ¼ inch thick it is the practice many times to have a 1/32-inch space between the punch and die at all points. The exact amount cannot be stated arbitrarily, it being governed by existing conditions.

There are instances in which it is advisable to make punches somewhat differently from the method described. When the nature of the stock to be punched is such as to cause it to cling to the punch, making the operation of stripping difficult, to the extent that any stripper plate put on the die would be bent, or the end of the punch pulled off during the operation, the punch may be made straight for a distance that allows of grinding several times; then the portion immediately above this may be given a taper. This tapered portion of the punch is intended to enter the stock, *but not the die*. Its action is to increase the size of the opening somewhat, thus making the operation of stripping possible without endangering either the stripper or the punch.

Advisability of Hardening Punches

There are various opinions among practical men as to the advisability of hardening punches. For most jobs it is the custom to do so, though there are some mechanics who consider it advisable to harden them, and others who do not. There are instances where punches work well either way, and in such cases it is, of course, a matter of opinion. If good results follow the use of a soft punch it may be used, and as the punch wears, it is upset and sheared into the die.

There are times when a soft die and hardened punch work well, and times when a hardened die and soft punch give good results. At other times both punch and die may be left soft. Very large punches and dies for hot trimming of drop forgings are sometimes used, where both are in a soft condition, and they stand up properly. The shape of these, together with the size, often make it impracticable to harden them. It would not be advisable to state that such and such dies or punches should be hard or soft; it must be determined by the circumstances under which they are to be used, and the decision is a matter of experience on that particular work.

Directions for Hardening Punches

If punches are to be hardened—and it is generally considered best—they should be very carefully heated. It must be borne in mind that punches are subjected to great strain, consequently they should be heated uniformly, and to as low a temperature as will give desired results, thus making them as strong as possible. Heat slowly to avoid *overheating* the corners, as these are subjected to the greatest strain. The distance we should harden a punch depends on the shape and size, and the use to which it is to be put. If it is a piercing punch of the form shown in Fig. 37, it should be hardened the entire length of the portion marked *a* to avoid any tendency to bend or upset when in use. If it is of a form that insures sufficient strength to resist any tendency to upset when in use, as is the punch illustrated in Fig. 38, then it need not be hardened its entire length.

Pack-hardening makes an admirable method for hardening punches for most work, but for piercing punches of the type in Fig. 37 it is not advocated, as the whole structure of the steel should be as nearly as possible alike. Such punches should be heated in a muffle furnace, or in a tube in the open fire, turning occasionally to insure uniform results, for not only can we heat a piece more uniformly if it is turned several times while heating, but a fact not generally known is that a cylindrical piece of steel heated in an ordinary fire without turning

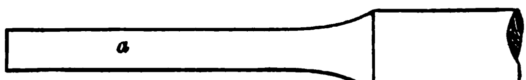


FIG. 37

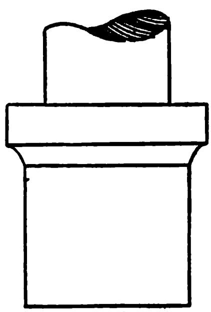


FIG. 38

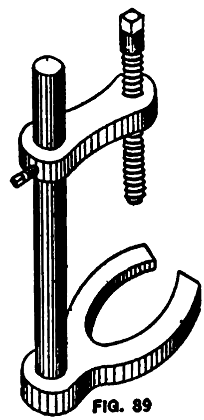


FIG. 39

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Figs. 37 and 38. Shapes Requiring Different Treatment in Hardening.
Fig. 39. Clamp Used when Scribing Die Outline on Punch

while heating will many times show softness on the side that was uppermost in the fire, no matter what care was taken when heating and dipping. If it is reheated with the opposite side uppermost, *that* will be found soft if tested after hardening, while the side that was soft before will be *hard*. The smaller the punch the more attention should be given to the condition of the bath. Luke warm brine is the best. Work the punch up and down and around well in the bath.

Tempering Punches

It is the custom of many mechanics to draw the temper of punches of the description shown in Fig. 37, to a full straw on the cutting end, but to have the temper lower further up the punch. Better results follow, however, if the punch is left of a uniform hardness its entire length of slender portion, as it is then of a uniform stiffness, and the liability of springing, especially when punching stiff or heavy stock, is reduced to a minimum.

It is generally considered good practice to temper the punch so that it is somewhat softer than the die; then, if from any accident the two

come in contact, the die will in all probability cut the punch without much injury to itself. There are exceptions to this, however. In many shops were large numbers of dies which are hardened are used, it is customary to have the one which is the more difficult to make the harder; so it will cut the other if they come in contact with each other.

In order to hold the die and punch blank firmly together when

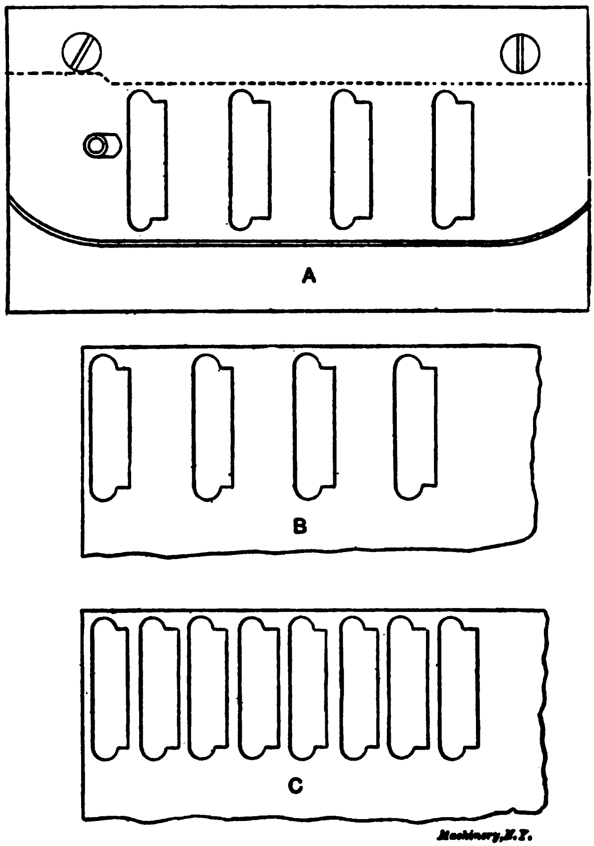


Fig. 40. Multiple Die, and Stock Cut in Same

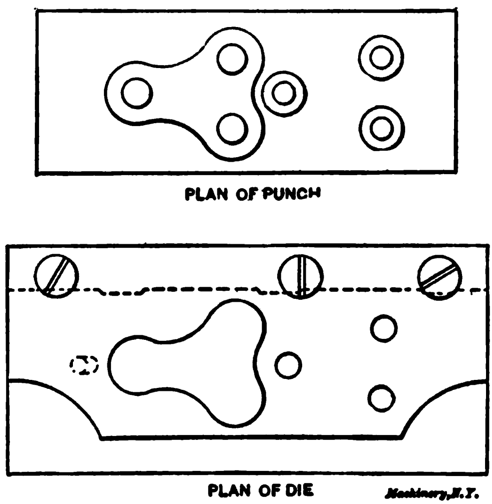
marking the shape on the face of the punch, a very convenient fixture known as a die clamp, shown in Fig. 39, is used. When the two are secured by means of this clamp, it is possible to move them around so as to get at the various portions where we wish to scribe.

Multiple Dies

A reduction in the cost of manufacture is often made possible by the use of multiple dies, whereby two or more pieces are punched out at a time. In punching perforated steel work it is no uncommon thing

to see punches and dies in use where several hundred punches are working into one die.

If an article, for example, of the form shown in the die in Fig. 40, were to be punched in lots of several thousand, the die should punch a number at a stroke. Such a die and the stock left are shown in Fig. 40, where the die is shown at *A* and the stock after the first punching at *B*. It will be noticed that the distance between the openings is considerable. This is necessary, as it would not be possible to place the openings in the die as close as they should be to econo-



Figs. 41 and 42. Gang Punch and Die

mize stock, since there would not be stock enough between to insure the die sufficient strength to stand up when working. For this reason the openings are located as shown. After punching as shown at *B*, the stock is moved along the right distance for the intervening stock to be punched out, as at *C*.

Gang Dies

If it were desirable to punch a piece like that at *a* in Fig. 43, it would be possible to make a blanking die and punch which would produce the blank of the right size and shape, but without the holes; then, by means of another die, with three punches working into it, we could punch the holes. It is apparent that such a method would be more expensive than one that made it possible to punch the holes and the piece at one passage of the stock across the die. This may be done by the use of a die of the description shown in Figs. 41, 42 and 43. When using this die the stock is placed against the guide and just far enough to the left so that the large punch *b* will trim the end. Then, when placed against the stop or gage pin *c*, bring the guide pins in end of punch *b* in line with the holes punched at the first stroke of the press at the time the end was trimmed.

When the stock is purchased of the proper width for one piece, it is fed through and the scrap thrown aside. At times it is purchased just wide enough for two pieces, in which case one edge is placed against the guide *d* and the stock fed through; after which it is turned over and fed through with the opposite edge against the guide, thus using all the stock except such portion as necessarily becomes scrap.

However, if the stock is purchased in the commercial sheet, it is necessary to trim the edges every time a row is punched along each. If no power shears are located handy to the press this may prove to be a more costly operation than the punching, and no matter how conveniently such a shear may be located, the operation adds a con-

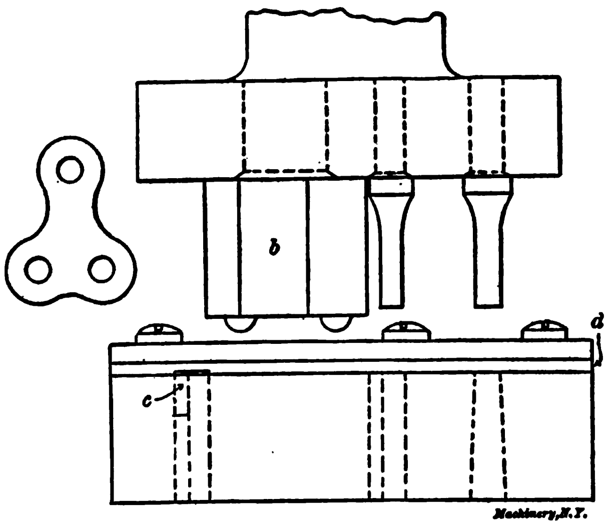


Fig. 43. Elevation of Gang Punch and Die shown in Plan in Figs. 41 and 42

siderable cost to the product. To avoid this trouble and expense another punch and opening in the die may be added. The object of this punch is to remove the scrap between the openings in this sheet and also trim the edge of the sheet, thus making it straight and in condition to bear against the guide on the die. The die and punch with the addition mentioned are shown in Fig. 44. When using a trimming punch as described above, it is necessary to use a stop of the description shown at *b*. The end of the scrap striking this governs the location of the stock, and when the punch descends the scrap is cut away.

When making dies of this class it is necessary to have the blanking die *a* the longer in order that the locating pins on the end may engage in the holes in the stock and locate it right before the other punches reach the stock. It is also necessary to place the gage pin, so that the stock will go a trifle further than its proper location—say 0.010 inch. Then, when the locating pins engage with the holes, they draw

the stock back to its proper location; whereas if the tool-maker attempted to locate the stop exactly, any dirt or other foreign substance getting between the end of the scrap and the stop would cause trouble.

Bending Dies

While it is possible, in certain cases, to bend articles during the operation of punching, it is usually necessary to make a separate operation of bending. There are instances where bending fixtures which may be held in a bench vise, or attached to the bench, answer the

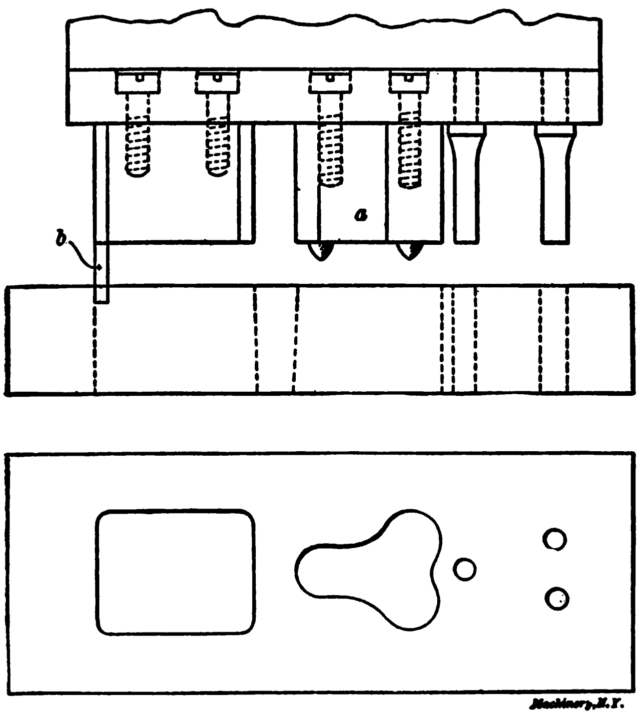


Fig. 44. Gang Punch Arranged to Use Sheet Stock

purpose as well and allow the work to be done more cheaply than if bending dies were used. But as a rule the die used in a press provides the more satisfactory method, and allows the work to be done at a fraction of the cost.

It is sometimes possible to make the dies so that the various operations can be done in different portions of the same die block, the piece of work being changed from one portion to another in order as the various operations are gone through. At other times it is necessary to make several sets of bending dies, the number depending on the number of operations necessary. When a "batch" of work has been run

through the first die, it is removed from the press and the next in order placed in, so continuing until the work has been brought to the desired shape.

When a comparatively small number of pieces are to be bent to a shape that would require a complicated and consequently costly die in order that the work might be done at one operation, it is sometimes considered advisable to make two dies, which are simple in form and inexpensive to make, to do the work. At times the design of the press is such that a complicated die could not be used; and as a result additional dies of a simpler form, and which can be fitted in the press, must be made.

We will first consider the simpler forms of bending dies. Fig. 45 represents a die used in bending a piece of steel, *A*, to a V-shape, as at *B*. In the case of a die of this form it is necessary to provide an

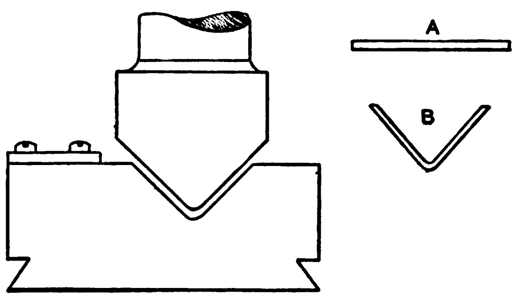


FIG. 45

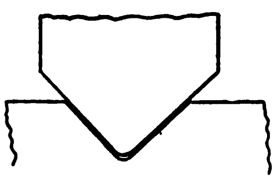


FIG. 46

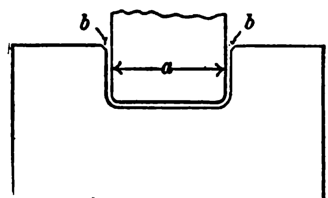


FIG. 47 Machinery,N.Y.

Figs. 45, 46 and 47. Examples of Bending Dies

impression of the proper shape as shown; this impression, if the die is to be used for bending stiff stock, must be of a more acute angle than if stock having little tendency to spring back when bent to shape be used. Under ordinary circumstances the upper portion or punch would be made of the same angle as the die. It is necessary to provide guides and stops as shown to locate the work properly.

If the stock used in making the pieces is of a high grade and the product is a spring or similar article which must be hardened, it will be found necessary to cut away the die somewhat in the bottom of the impression, making it a little different in shape from the punch, as shown in Fig. 46. This is to prevent crushing or disarranging the grain of the steel to an extent that would cause it to break when in use.

If the die is of the form shown in Fig. 47, it is, of course, necessary to make the length *a* of the punch shorter than the distance across the opening of the die. It must be somewhat shorter on each end than the thickness of the stock being worked. If possible, the upper corners *b b* of the die should be rounded somewhat, as the stock bends so much easier and with less danger of mutilating the surface than when the corners are sharp. When bending thin ductile metal the corners need but little rounding. If the stock is thick, or very stiff, a greater amount of rounding is needed.

While the form of bending die in Fig. 45 answers for ordinary work, there are jobs where such a die would not insure a degree of accuracy that would answer the purpose, and it will be found necessary to make one similar to Fig. 48, where a riser or pad *a* is provided, as shown.

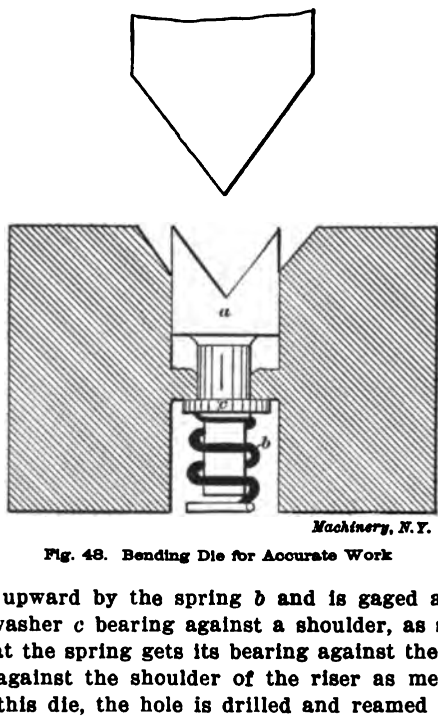


Fig. 48. Bending Die for Accurate Work

This is forced upward by the spring *b* and is gaged as to height by means of the washer *c* bearing against a shoulder, as shown. It will be observed that the spring gets its bearing against the washer, which in turn bears against the shoulder of the riser as mentioned before. When making this die, the hole is drilled and reamed and the groove milled or planed for the riser, which is put in place sufficiently tight to hold it while the V-groove is cut, after which it may be relieved until it works freely. The spring *b* gets its lower bearing on the die holder. If it is considered advisable, a screw may be provided for the spring to rest on. By adjusting this screw, any desired tension may be given the spring, although, generally speaking, this is not necessary.

When bending articles of certain shapes it is necessary to design the tools so that certain portions of the piece will be bent before other portions. Should we attempt to make the tools solid and do the work

at one stroke of the press, the piece of stock would be held rigidly at certain points and it would be necessary to stretch the stock in order to make it conform to other portions of the die. In the case of articles made from soft stock, this might be accomplished, but the stock would be thinner and narrower where it stretched. However, as a rule it is not advisable to do this, and dies are constructed to do away with this trouble.

Fig. 49 represents a die, the upper part of which has the portion *a* so constructed that it engages the stock first. After forcing it down into the impression in the lower portion, part *a* recedes into the slot

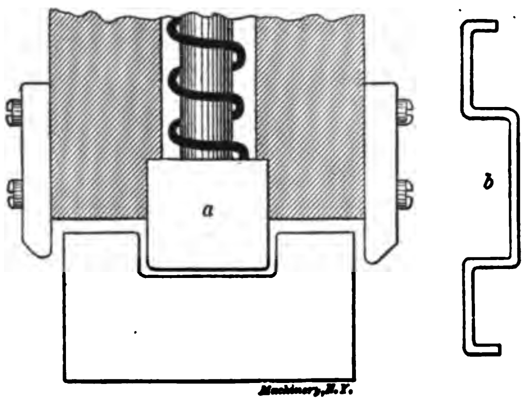


Fig. 49. An Example of Progressive Bending Die

provided for it. The coil spring shown is sufficiently strong to overcome the resistance of the stock until it strikes the bottom of the impression. The article is shown bent at *b*.

Compound Bending Dies

Compound bending dies are used very extensively on certain classes of work, especially in making looped wire connections and articles of thin sheet stock. Fig. 50 shows a die used for bending a bow spring. As the punch descends, the stock is bent down into the impression in the lower half and forms the stock to a U-shape. As the end of the punch with the stock comes in contact with the bottom of the impression it is forced into the upper portion, the spring keeping it against the stock, while movable slides—side benders—*b* are pressed in by means of the wedge-shaped pins so as to force the upper ends of the loop against the sides of the punch as shown in Fig. 51, forming the piece as at *B*. When the punch ascends, the finished loop may be drawn off. If the stock used is stiff it will be necessary to make the punch somewhat smaller than the finished size of the spring, as it will open out somewhat when the pressure is removed.

When making looped wire work, a loop may be formed and the wire moved along against a stop, another loop formed, and so on, as in Fig. 52. When forming looped wire work it is customary to make the

punch ball-shaped rather than as shown in Fig. 50. The ball answers well on wire work and allows of the easy removal of the loop. It is sometimes desirable to close the upper end of an article nearly together, and if the stock used is extremely stiff, as bow springs made from a grade of tool or spring steel, it may be necessary to heat the

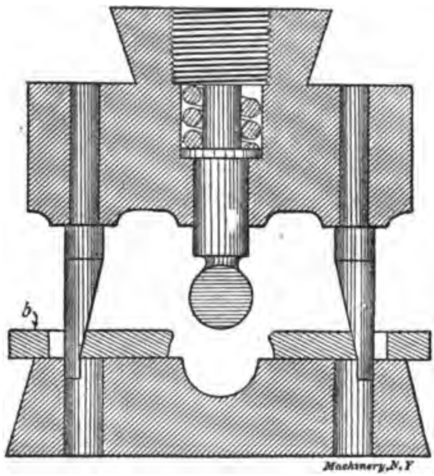


Fig. 50. Die for Bending Bow Springs

bow, which has previously been bent, red hot, and finish bend it by a special process. In the case of articles made from a mild grade of stock the whole bending process may be accomplished in one operation by substituting a mandrel, as shown in Fig. 53, for the cylindrical portion of the punch.

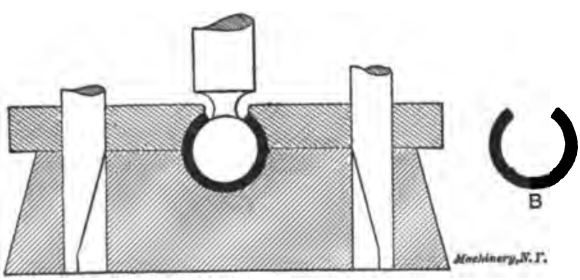


Fig. 51. Action of Die in Fig. 50

A great variety of work may be done by modifications of the methods for bending shown. Where but a *few* pieces are to be bent it is not advisable to go to the expense of costly bending dies; but when the work is done in great quantities, they will produce work uniform in shape at a low cost. Blanking and bending dies are made which not only punch the article from the commercial sheet, but bend it to the desired shape at the same operation. As a rule, it is advisable to

blank the article at one operation and bend it at another, but there are certain forms of work where it is possible to do it in a satisfactory manner at one operation and at a cost not exceeding that of the ordi-

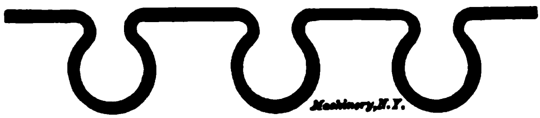


Fig. 52. Successive Loops Formed in a Wire

nary blanking operation. This also effects a saving in the cost of tools, as the special bending die is dispensed with.

Fig. 54 represents a punch and die used in punching the shoe *a* to the proper shape shown, while Fig. 55 is one used for producing

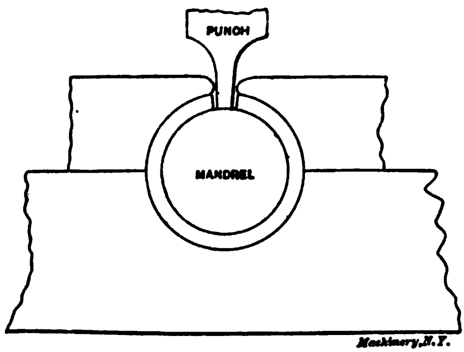


Fig. 53. Forming a Bow Spring with Ends which nearly meet

the tension washer shown. Gun and other irregular shaped springs are many times punched to form by this style of die, although, when stock suitable for use in making springs is employed, it will be found necessary to make the face of the punch somewhat different in shape from

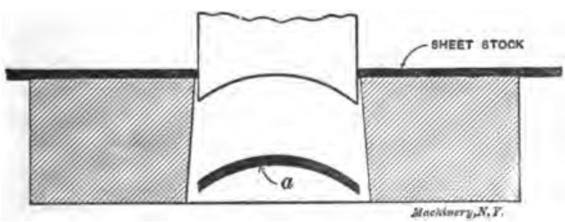


Fig. 54. Punching and Bending at One Operation

that desired, as the piece will straighten out more or less after it is punched.

If it is desired to curl a form on a piece of work, making a loop as in Fig. 56, it is accomplished by various methods, sometimes by a modification of the die in Fig. 51. A die of the description shown in Fig. 57 is used with excellent results. In making this die, the blank *a* is first machined to size. The hole *b* is drilled and reamed to size,

and polished to produce very smooth walls. This may be accomplished by using a round revolving lap of the right size. The slot is then milled as shown. If the die is not intended for permanent use and the stock is comparatively soft or easily bent, it need not be hard-

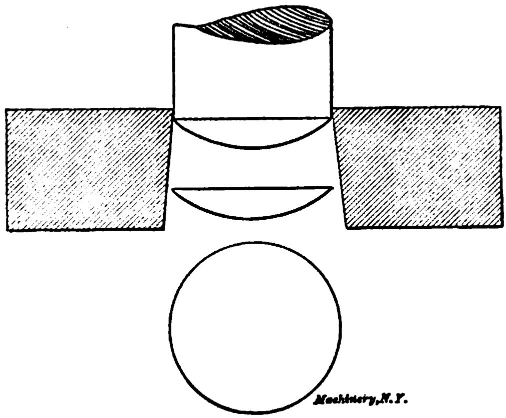
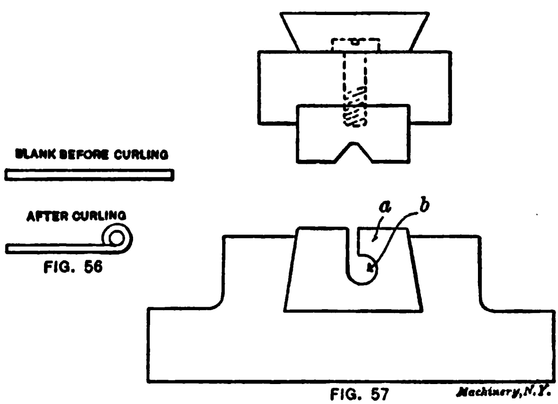


Fig. 55. Making a Tension Washer

ened. If, however, it is to be used right along, it must be hardened. This is best accomplished by pack-hardening, being sure that the heat is low. As when using this method the die is quenched in oil, there is little or no danger of its going out of shape. It is then drawn



Figs. 56 and 57. A Curling Die and its Work

to a full straw color. The punch is made with a V-shaped impression in its face, as shown. This may be flat in the bottom, as indicated, or left sharp, as desired.

It is possible with presses and tools adapted to the work to form pieces to shapes that to one not familiar with this class of work would seem well-nigh impossible.

CHAPTER II

SUGGESTIONS FOR THE MAKING AND USE OF DIES

In the phenomenally rapid progress made during the last decade in the press working of sheet metals by the introduction of compound, combination, sub-press, and gang dies, automatic roller and dial feeds, the simpler operations on the power press, instead of becoming subject to similar improvement, have been sadly neglected. It is therefore not out of place to refer, shortly, to the basic elements of the art of using and making dies. Although the following discussion originally was intended to apply to one particular line of presses, the suggestions brought forward may be applied with slight modifications to any make of upright power press on the market to-day.

It is not so generally known as it should be that the inclining of a press adds materially to its productive capacity. This advantage is almost doubled when the same belt may be used in both positions, permitting the change to be readily made without undue loss of time. Many users make it a rule to incline the press on all operations except "push through" jobs, that is, on all work which does not drop through the bed of the press. It is then simply necessary to feed the work to the dies, allowing it to drop out by gravity. To permit the use of the same belt for both positions, the press should be so placed on the floor that the center of the shaft when in its inclined position is the same distance from the line shaft as it is when the press is upright.

While there are many diemakers who advocate the use of a separate cast iron bolster for each die, it is advantageous to use bolsters made of cast steel, which are largely used by Western shops. There are two made for each press, one for cutting dies and one for bending and forming dies, the construction of compound and combination dies remaining unchanged. By this system the separate dies are interchangeable on any press; they occupy less space on the shelves of the tool-room, and inasmuch as all strippers and gages are fastened directly to the die instead of to the bolster, they never become lost when changing from one job to another. The desirability of using standard hexagon head cap screws to hold down strippers, gages, etc., should be impressed upon diemakers. The strippers on any die may then be removed to facilitate correct setting of the die, and then replaced in position—something impossible on slotted head screws except by using an angle screw-driver.

There is little room for improvement in the cast iron punch-holder. One might suggest, however, the use of solid piercing punches in place of the drill-rod surrounded by a soft steel sleeve riveted to a punch-pad. Wherever possible, it is advisable to do away with the old-fashioned soft steel punch sleeve, and to let the punches into their holders

either by turning a round shank on them, or dove-tailing them into the cast iron holder in the same manner as the die.

In planing up the die-blank it is well to remember to take a very slight cut from the bottom and a cut about twice as deep from the top.

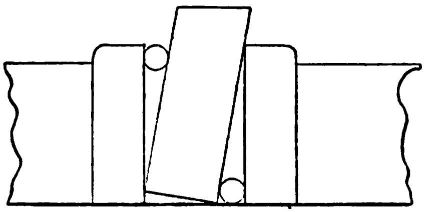


Fig. 58

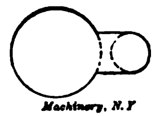


Fig. 59

This removes the decarbonized surface from the cutting face where it needs most to be done, but leaves it on the bottom where the die may remain soft. Where there is a scarcity of 10-degree parallels, two pieces of drill-rod between the jaws of the vise may be arranged to

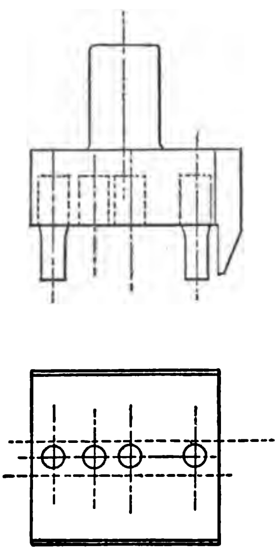


Fig. 60

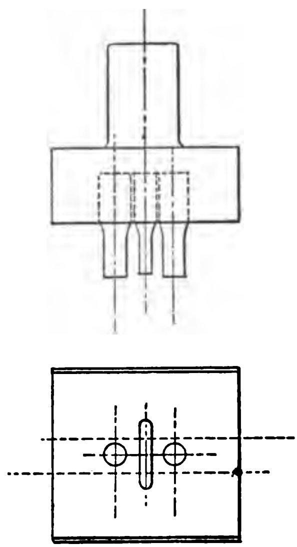


Fig. 61

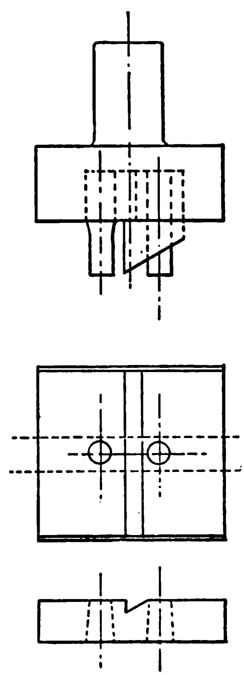


Fig. 62

Steps in the Evolution of Press Tools for Copper Connectors shown in Fig. 65

give the correct angle, as shown in Fig. 58. Quarter-inch drill-rod is the size to use when the jaws are 1 9/16 inch high. Where intricate shapes must be drilled out with small drills, the holes may be laid out a trifle close together, and the shank of an old drill of the same size

pushed into the first hole drilled. This will prevent the drill from running too far into the previously drilled hole, and by proceeding in this manner all around the outline, the core to be removed will drop out without the use of chisel or drift. The amount of draft on some blanking dies which are combinations of drilled holes, as, for instance, the shape in Fig. 59, may be infallibly obtained by reaming these holes from the back of the die as though they were simple piercing dies. Where extreme accuracy is essential, or a die is too large to be made of a single forging, the use of sectional dies becomes imperative. While the first cost of a well-made die of this kind is higher than that of a solid die, still the ease of repair and uniformity of production of this type of die make it advantageous in the long run.

The dies shown in the illustrations serve to emphasize the main features of this discussion. Fig. 60 shows a die as originally made for the three copper connectors shown in Fig. 65. It is a plain cutting-off die, having the different holes placed in the die at the proper center distances apart. By means of a suitable adjustable gage and by placing one of the piercing punches in its proper position in the punch-holder, the three different sizes of connectors shown in Fig. 65

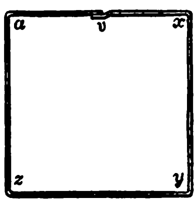


Fig. 63

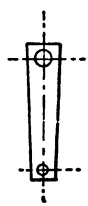


Fig. 64

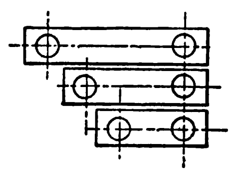


Fig. 65

may be produced. However, during the process of improvement of the device on which these connectors were used, it became necessary to change the center distances between the holes and also to produce three longer ones. The die shown in Fig. 61 was at first considered adequate, but, on account of the quantity required, the scrap produced by the cutting punch was considered objectionable. Leaving the piercing punches in the same position, the shape of the cutting-off punch was changed, as shown in Fig. 62, and a corresponding V-groove planed in the die. In connection with stripper and gage (not shown) this die allows the production of an indefinite number of connectors of different center distances.

The die shown in Fig. 66 impresses the fact that the slitting shear is a valuable auxiliary to any press. The metal for the production of the copper segment ⅛ inch thick, shown in Fig. 64, ordinarily would be cut a little wider than the length of the blank so as to allow the punch to cut all around. But in all cases where at least two sides of a blank are parallel, the stock may be cut the exact width of the parallel portion of the blank in the slitting shear, and then the pieces may be punched and cut off two at each stroke of the press, as shown in the die in Fig. 67. There is one inherent drawback to this form of die,

and that is the tendency of the punch to lift up the end blank while cutting it off and produce a badly beveled edge. But if this portion of the strip is securely held down by the clamping device on the die as shown, the punch will have the same effect on both sides of the blank, cutting it off squarely. The gage and stripper held down by the cap-screws can be made a better fit on the stock than ordinarily, because it is not necessary to lift it up past a stop pin fastened to the die to enable the operator to feed the strip. By inclining the press, allowing one blank to slide out when released by the clamp, and letting the punched one drop through, two complete blanks are produced at each stroke of the press, with almost no scrap.

The extension punch and die in Fig. 67 is quite useful on work which is commonly beyond the scope of the press, such as the sheet

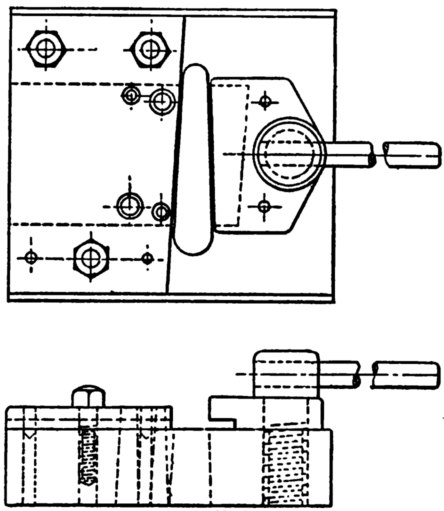


Fig. 66. Die for Punching without Waste the Pieces shown in Fig. 64

iron box shown in Fig. 63. This forms the sides of a slate-bottomed switch cabinet used on the old Manhattan Railway cars when they were equipped with electricity. The operations on this box included the bending of the 2 by ⅛-inch strap iron in four places, forming the lap joint, and riveting same. The cut shows the punch and die (without necessary stops and gages) in position for bending the corners. The front clamping plate is removed from the ram and a cast steel extension bolted in its place with the same bolts. The large hook bolt extending into the hole in the ram and drawn up by the nut outside, is required to support the extension during the strain of bending. To allow the stock to clear the front of the press when bent into shape, the distance *A* in Fig. 67 should be a little more than half the width of the strap iron to be bent, and to avoid fouling the flywheel, corner *x* in Fig. 63 should be the first one bent after the lap

has been formed, and then, in rotation, corners *y*, *z* and *a*. When running the press at its accustomed speed on this job the ends of the bent piece moved rather too fast for comfort, and it was therefore necessary to cut down the speed of the flywheel by inserting resistance in the armature circuit of the motor which drove the line shaft to which three of these presses were belted.*

Method of Locating Stock in Dies

When a job will not warrant the expense of a sub-die, the device shown in Fig. 68 will help wonderfully toward producing accurate punchings. To simplify the explanation, the die shown is to cut washers, the holes being eccentric with the outside. The die is laid out the same as any double die, but the stop pin *G* is added, and as will be noted, the extension *K* does not come out of the die. If, however,

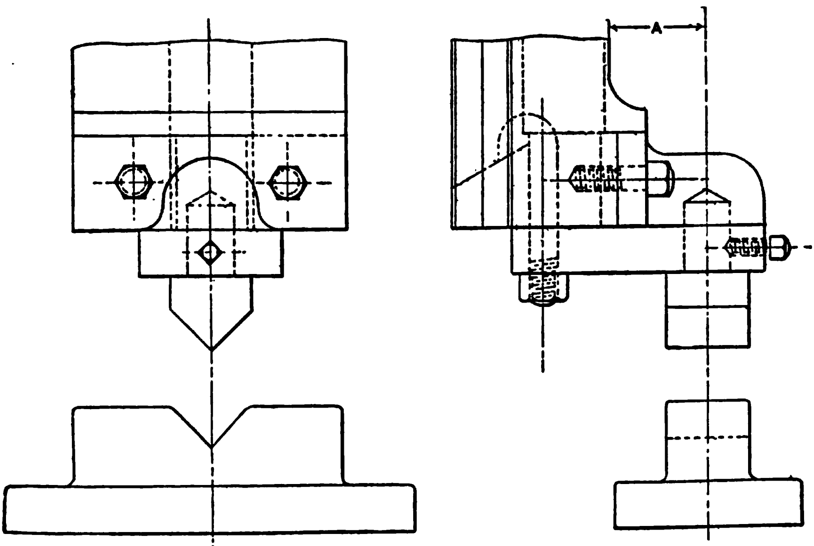


Fig. 67. Die for Corner of Sheet Iron Box

one depends entirely on this stop pin, the result will not be satisfactory, because, when the stock is pulled against the stop pin, the web between the blanked places will bend a trifle, especially if the stock is thin. Therefore the long pins *H* are added, and as these long pilots or traveling dowels are well pointed, and are considerably longer than the punches, they of course enter the holes and force the stock back to its proper location. The pilots fit two holes in the die, and they therefore act as dowels while the punch is cutting. The pilots and the spring butts *L* keep the stock pressed firmly against the gage side of the stripper, and the stock can vary 1/16 inch. With this construction the operator is enabled to keep the press running constantly to the end of the strip. At each stroke the punch *G* cuts out the web and allows

* H. J. Bachmann, July, 1906.

the stock to slide along to the next web, and there is absolutely no possibility of the stock jumping the stop.

As washer or small wheel dies are generally made to cut four or more blanks at one stroke, the following method of transferring the holes to stripper and punch-holder will be of benefit to some mechanics. If the punches are small, it is advisable to make the stripper, say, ½ inch thick, and dowel it with four good-sized pins to the die. The holes through the stripper are bored to fit the punches nicely. This will act as a guide and prevents the punches from shearing. When the stripper is doweled to the die, we lay out the former with buttons or by other methods governed by the accuracy demanded, and each hole in

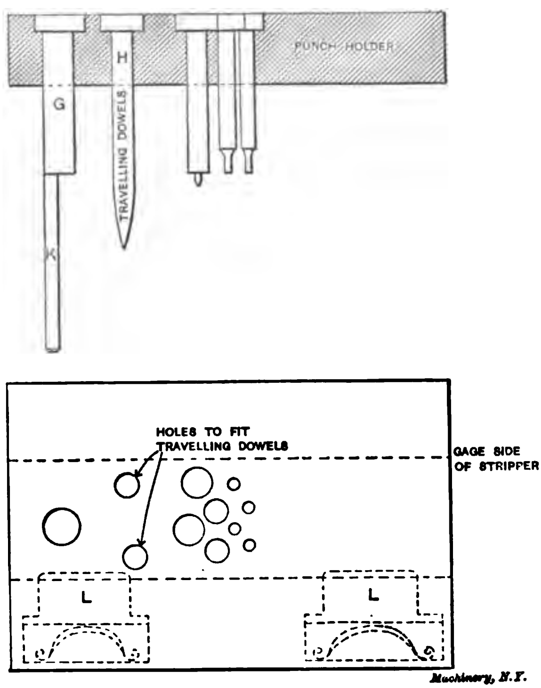


Fig. 68. Punch and Die with Guide Pins

turn is indicated and bored through the stripper and die. If the holes are so small that they will not readily admit boring to such length, the stripper may be bored and removed and the die then bored. The die must, of course, be fastened in such a manner that the stripper can be removed without loosening the die. If properly doweled, the punch-holder, stripper and die can be bored together, thus insuring perfect alignment of the punches and the die.

Making an Irregular-shaped Die

Fig. 69 shows a time-saver, as the die can be made easier and better because the parts can be ground to size instead of the die being filed

out. Another advantage is that if the pieces warp in hardening they can be ground into shape again. The pieces *M* are shrunk on the sections, holding them securely together. The holes *N* are drilled for clearance for the emery-wheel when grinding to size. The straps *M* are made a trifle shorter than the die over all, say 1/16 inch to the foot, and are heated red hot in the middle and placed in position while hot, and rapidly chilled. After these pieces are shrunk on, the dowels are transferred into the bolster.

Another good kink when making irregular-shaped punches that are to cut thin stock is to make them of machine steel and case-harden them. Soft steel, case-hardened, does not change its form as much as

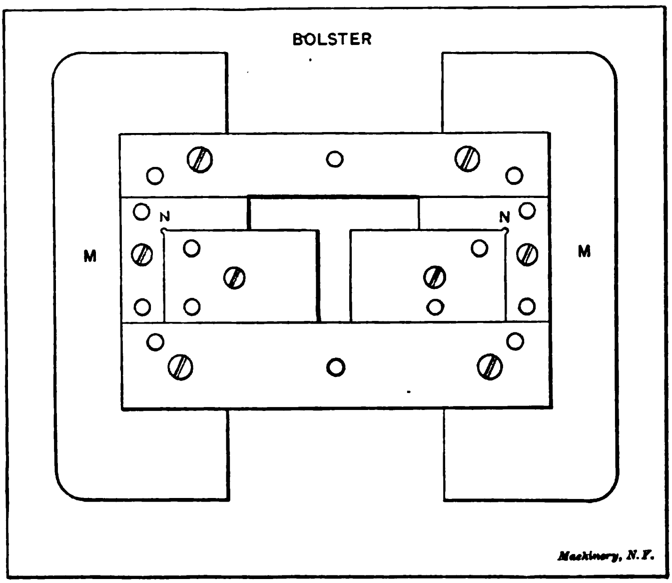


Fig. 69. Example of Built-up Die

tool steel, and even if the punch does change a trifle, the interior is soft and can be readily forced back to position. The outside being hard, the punch will wear nearly as long as one made from tool steel, for practically the only wear on a punch is when passing through the stock. For thin brass the punch works well when made of tool steel and left soft, and when worn badly the punch can be peened on the face enough to upset, and then sheared into the die. When cutting a heavy blank, it is a good plan to grind the die so that the surface is quite rough, as the high spots then cut a trifle ahead of the low points. This will cause the die to run longer between grindings and is also easier on the press, while with a die that is ground perfectly smooth the entire cutting surfaces of punch and die meet simultaneously and the entire cutting surface of punch and die are placed under a tremendous strain. By grinding the die slightly lower on each end, thus producing a shearing cut, the die will last longer.

A Kink in Hardening

What will greatly reduce the chances of springing in hardening of an irregularly shaped punch or die is to thoroughly anneal it after it has been machined nearly to size. This will, of course, not entirely remove chances of accidents, as the prime cause of cracks and distortion of work is to be found in the operator's way of handling the piece to be hardened. An illustration of what takes place when hardening may be given by referring to the die shown in Fig. 70. If we place the die in the fire, the points *C* will heat and expand quicker than the main body of the die, and there must be a sort of a "pushing" effect between the points *C* and the main body of the die. For this reason we heat "slowly and evenly." Now, when we dip the die in the bath, the points *C* immediately become chilled, and, of course, contract while the main body is still red hot. Assuming that the points have become entirely cooled, there must be a line that separates the part

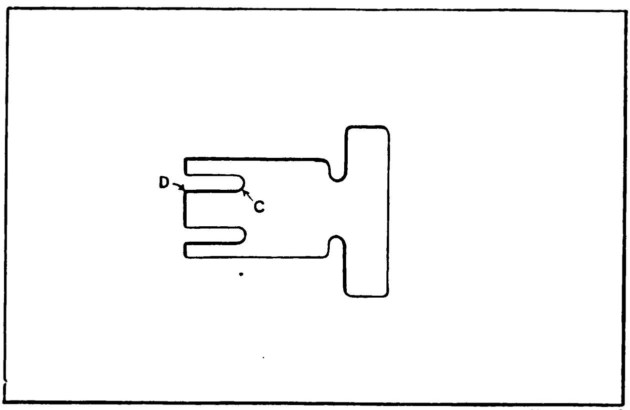


Fig. 70. Die of Irregular Shape Subjected to Heavy Strains in Hardening

that has been cooled off from the red-hot part. It must follow that when the main body begins to contract there is a powerful strain at the line that separates the parts contracting at different times. For this reason the die should be removed when quite warm; this allows the heat to run out into the points and the contraction will be more even. If allowed to cool in the bath there is apt to be a crack at *D*. Polish the die to draw the temper, and do not depend on getting an even temper by drawing the die when it is dirty, as one part may draw faster than another.

Doweling Hardened Parts

When making pieces such as sections of a built-up die, or any piece having dowel holes, it invariably happens that the dowel holes do not line up after hardening. One way to overcome this trouble is to tap the dowel holes a trifle larger than the dowels to be used, and after the piece is hardened, screw in soft plugs and file them off flush with

the work; when the piece is screwed in its proper place, the dowel holes are drilled and reamed through the soft screw bushings. This will save a great deal of unsatisfactory lapping.*

Construction of Dies to Prevent Breakage in Hardening

Another method of preventing breakage in hardening of dies with small projecting tongues, as shown in Fig. 70, is to construct the die in the manner outlined below. The die is first filed or machined in the regular way, with the exception that the two tongues are left out. In line with the center of the tongues and at a certain distance from the cutting edge, holes are drilled larger than the width of the tongues. These are taper reamed from the top with a standard taper reamer. A slot is then cut from the holes into the die the same size as the

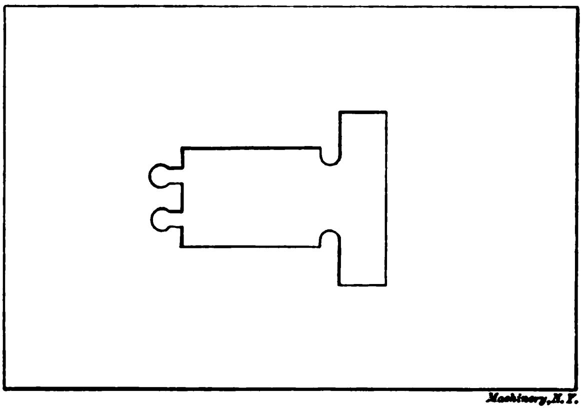


Fig. 71. Method of Making Dies to Prevent Breakage in Hardening

tongue, when the die would look as shown in Fig. 71. We now make two pieces to fit in the holes, and extend out the required distance, making sure that they will be a drive fit after hardening. It is best if the pieces are 1/32 inch longer than the thickness of the die, so that they can be ground flush after being driven into place. While this may increase the cost of producing the die, yet, if from any accident one or both tongues should be broken, they are easily replaced without the necessity of annealing the die.**

Fig. 72 shows a very good method of making a die that is to contain a number of identically shaped teeth or points, such as dies for gear blanks, etc. While not being the most accurate method known, it is considered that for all work intrusted to a punch and die the method illustrated will be sufficiently accurate. A set of broaches are made, as shown in the cut, the number of steps being governed entirely by the length, or depth, of the teeth. The pilot fits the hole in the die, which is the diameter at the top of the teeth, and each step on the broach is 0.002 inch larger than the preceding step. The broaches

* F. E. Shailor, March, 1907.
** K. L. Ross, September, 1907.

are made on centers and necked in at *Q* to allow clearance for the chips. With a cutter of the proper shape the teeth are then milled on the broaches, using the dividing head on the miller. After cutting the teeth on all of the broaches, the teeth on the punch should be cut

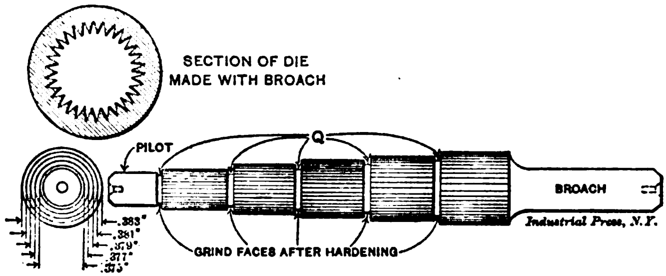


Fig. 72. Broach for making Dies for Gear Blanks, etc.

at the same setting. The broach is then hardened and ground on the faces as indicated. When used, each successive step is driven through the die until the last step is reached, and this should be driven through as many times as there are teeth in the broach, turning it one tooth each time. By doing this, whatever error may have been caused by hardening is overcome.*

* F. E. Shailor, January, 1904.

CHAPTER III

EXAMPLES OF DIES AND PUNCHES*

In the following are given a few examples of the design and construction of dies and punches, selected because they are very interesting and ingenious in their action. The die in Fig. 73 was designed by Mr. Thomas Gierding, of the New Haven Clock Company. This die performs five distinct operations before the piece shown in the upper left-hand corner of Fig. 73 is dropped completed from the press.

In constructing this die it was not deemed practicable to make it of one solid piece, since one small flaw would, in this case, spoil the entire die. A die block of machine steel was therefore used, having recesses counterbored for the insertion of tool steel bushings. These recesses were accurately spaced by the method illustrated in Fig. 74. One side and one end of the die block were machined perfectly square, and a center line drawn lengthwise on the face of the block. The location of the recesses was approximately laid out with lead pencil and the recess *A* bored in the lathe, by strapping the block to the faceplate. Before loosening the straps by which the block was held, the parallels, *B* and *C*, bearing against the finished edges of the block, were strapped to the faceplate. The straps holding the block were then loosened and the block moved along the strip *C* sufficiently to allow for the insertion of the spacing block, *D*, which had previously been made of the required size. The die block was then fastened and the hole *E* recessed. By repeating this operation, and adding a block each time until all of the recesses were bored, it was possible to space the die far more accurately than would have been possible by the time-honored method of laying it out with dividers. The punch-holder and the stripper were then bored in the same manner, using the same spacing blocks.

The bushings *F*, *G*, *H*, *I*, *J*, *K*, were next made, and after being hardened they were lapped to size. The outside of the bushings was ground concentric with the hole by wringing the bushing on a piece of soft steel held in the chuck and turned to fit the hole in the bushing. The bushings were then forced in the die block and the die was completed. The punches were ground all over, to insure straightness, and they, in turn, were forced into the punch-holder. The drawing and forming punches *L* and *M* were held with setscrews to prevent them from being pulled out.

In using a die containing two or more punches, considerable trouble is sometimes experienced on account of the variation in width of the stock to be punched. Should the stripper be planed to fit one of the strips of stock very nicely, the chances are that the next strip would not enter the stripper at all. The part, *N*, shown in the plan of the

* MACHINERY, January and August, 1904.

die, is a novel and practical way in which this trouble is overcome. The stripper is planed out 1/16 inch wider than the stock and recessed to allow the spring guide *N* to slide freely when the stripper is in

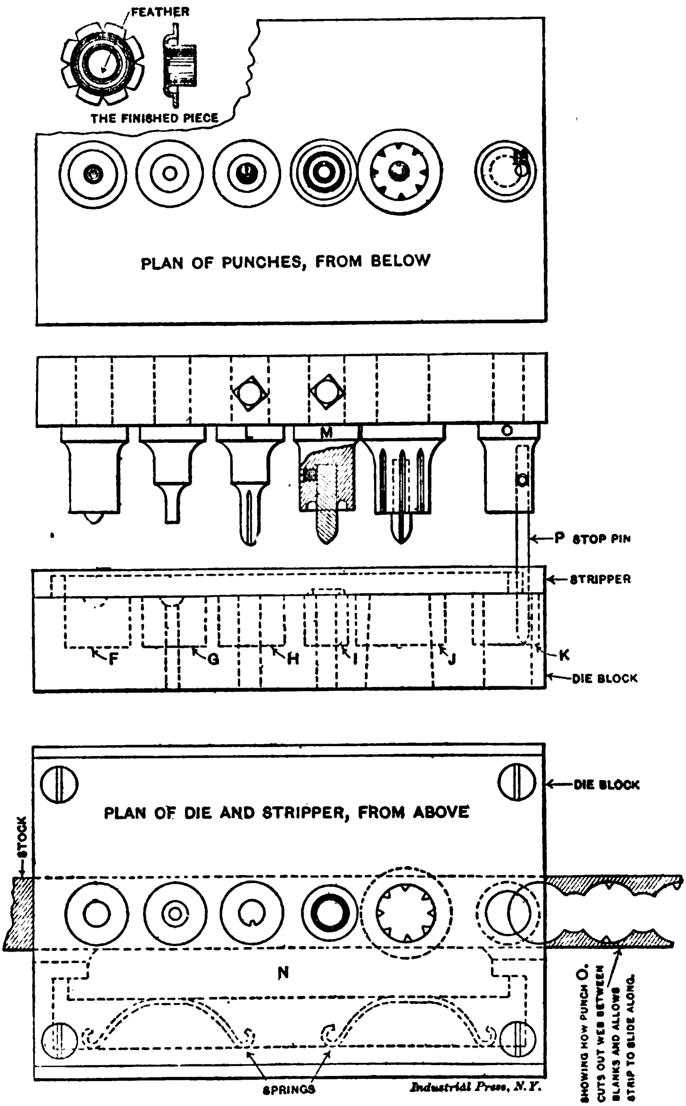


Fig. 73. Punch and Die for Performing Five Distinct Operations

position in the die. By glancing at the sketch the reader can readily see how the springs keep the stock pressing against the gage side of the stripper. The punch *O* does not perform any work pertaining to

the finished blank, but is used for cutting out the web in the stock in order to allow the strip to move along until the next web touches the stop pin. As the stop pin *P* does not come out of the stock it is therefore impossible to "jump" the stock and make a miscut, which would mean disaster to the drawing and forming punches.

After setting up the die in the press, the punches of course descend five times before a single finished piece appears, but thereafter a finished piece drops at each stroke of the press. The first punch, beginning at the left, indents the stock, and the punch is so adjusted that the face of the punch levels the stock. The second punch pierces the bottom of the indentation. The next punch draws the stock, and at the same time, forms the feather shown in the finished piece. The fourth is the forming punch and the last punch does the blanking.

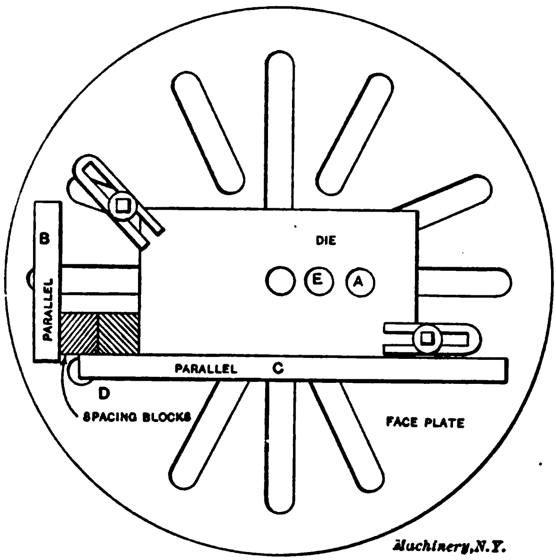


Fig. 74. Spacing the Holes in the Die in Fig. 73

Another interesting die is shown in Fig. 75. This die contains several novel features that will be found valuable to many engaged in die-making. As the sub-press die, the frames, and the power presses are of standard dimensions, it too frequently occurs that a die of a certain size requires specially made frames, and possibly a specially made press. The cut, Fig. 75, shows a practical way to construct a die that not only is a compact self-contained die, but that can be fitted to any style of press (of sufficient strength), having any length of stroke.

This particular die was designed to produce the disk shown at *A*, Fig. 76, and previous to its introduction the disks were blanked out with a plain open die and then leveled by hand. The disks are of aluminum, 99 per cent pure, and, therefore, very soft, and as it is very essential that they should run as true as possible, great difficulty was

experienced in leveling them. The corrugating mats *B B* were designed to level the disk and also to set, or stiffen the metal, and they proved a success, for when the disks leave the die they are as nearly level and true as is possible to make, and so stiff that they can be handled quite roughly without injury. The disk was not corrugated its entire surface owing to the fact that the mat would be obliged to act as the blanking punch, and if the corrugations extended clear to the edge of the mat, it could not be sharpened when dull. Therefore the rings *C D* were introduced. The ring *C* acts as the blanking punch, and ring *D* acts as a leveling ring. The die is guided by means of two guide or pilot pins, *E E*, Fig. 76, and as the gate of the press descends, the rings *C D* are the first to act on the stock to be punched, gripping it from above and below and holding the stock securely. Then, as the press continues downward, the rings settle back, still holding the stock, and the mats *B B* grip the blank.

The rubber spring, which is one of the features of the design, exerts an increasing pressure on the metal, pressing it into the corrugations

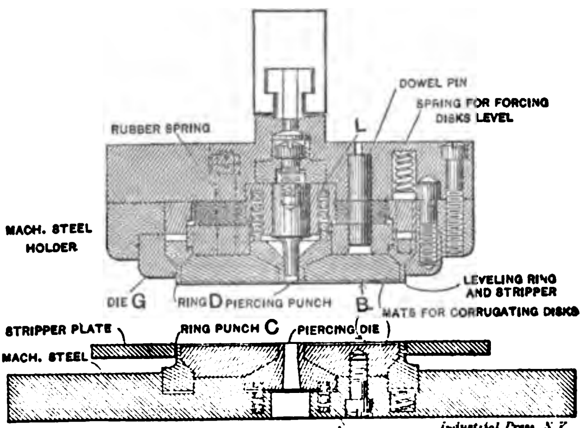


Fig. 75. Vertical Section through Sub-press Die shown in Fig. 76

on the mats. The press is so adjusted that the ring *C*, which is the blanking punch, comes exactly flush with the die *G*, but does not enter. On the upward stroke of the press, the springs and rubber plate force the moving parts back to their original position, and force the disk out of the die, and the surplus stock off the ring *C*. The rubber plate can be advantageously used in a small place where a very strong spring is required. The tension or spring effect is obtained by cutting holes *H* in the plate, Fig. 77. The more holes there are in the plate, the weaker the tension, as the holes permit the surrounding rubber to squeeze into them. On the other hand if no holes were cut in the rubber plate, and the same fitted the recess in the die bored for it, there would be no more spring effect than if a metal plate were used. Rubber does not compress, but merely changes shape. Another novel feature is

that the guide pins are automatically lubricated at each stroke of the press. The pins run in the babbitt boxes *I I*, Fig. 76, which have four grooves, *J*, cut the entire length of the babbitt and an oil chamber or reservoir *K* recessed near the top. A quantity of oil is placed in the bottom of the box and as the pins descend they force the oil up through the grooves, *J*, into the reservoir, and as the pins ascend they form a partial vacuum at the bottom of the box, which sucks the oil back to the bottom.

Space will not allow describing the methods employed when making each part of the die, but it will suffice to say that with the exception

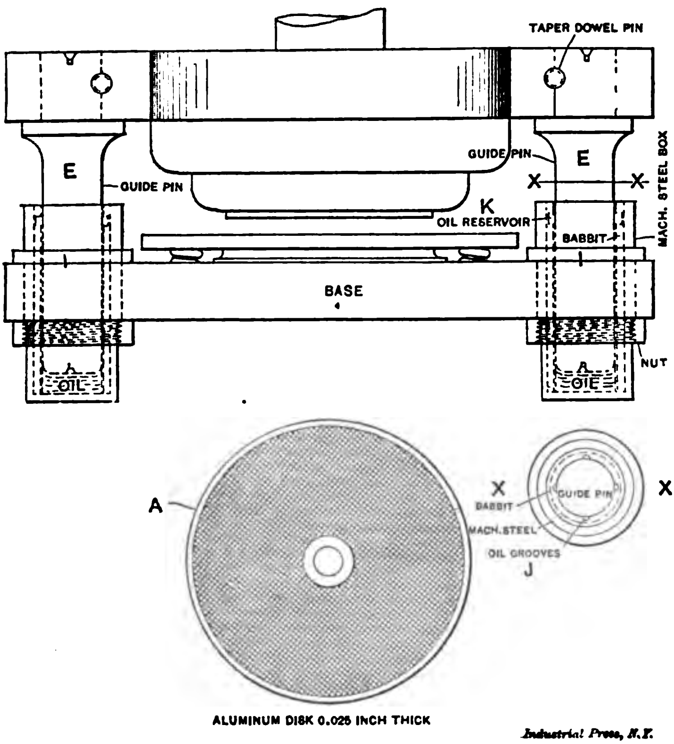


Fig. 76. Side Elevation of Die shown in Section in Fig. 75, and Sample of Work

of the mats, screws and holder, the parts were hardened and accurately ground, making a smoothly running die. It might be well, however, to mention the method employed in making the square springs *L*.

It is well known what a difficult job it is to wind a heavy coil spring and have it a given diameter on the inside and outside, when finished. A large spring is generally made by heating wire red hot, and winding as many coils as possible before cooling, then reheating and winding more coils. The springs *L L* were made by gripping a piece of round

tool steel in the lathe chuck, turning it to the given outside diameter. The lathe was then geared to cut a coarse pitch thread and with a square thread tool, the thread was cut sufficiently deep. The inside of the spring-to-be was then bored out to the proper diameter, leaving a spring the coils of which were evenly spaced, thereby causing each coil to perform equally its share of the work. With a wound spring the coils are very seldom equally spaced, and when under pressure there is a greater strain on the coils furthest apart, causing the spring to either "set" or break at that point.

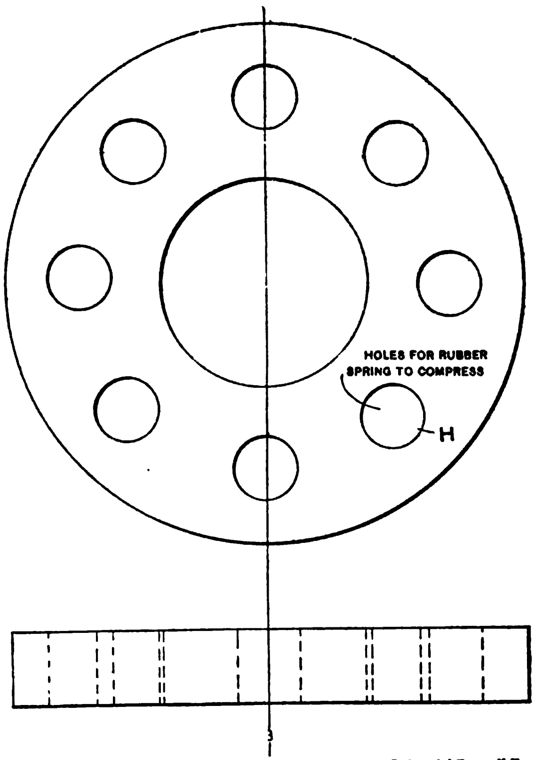


Fig. 77. Spring Rubber Plate

After all parts of the die were completed, the die was assembled, leaving out the springs. The upper and lower parts were then brought together until the punches entered the dies, care being exercised that the upper and lower parts of the die were perfectly parallel with each other. The boxes *I I* were then babbitted, first treating the guide pins with a light coating of flake graphite and oil to prevent the babbitt sticking to the pins. The writer considers that a large die of the above description is far superior to the ordinary sub-press die, inasmuch as it

is more compact, and also does away entirely with the cumbersome cast iron frame.

Fig. 78 shows a die that is designed to take the place of the plain, open, double die. The ordinary double die is made with the stripper fastened to the die and planed out to allow the stock to slide through. The unsatisfactory results obtained when using a die of this style are well known. The greatest fault is that no two blanks are exactly alike, owing to the fact that the stock is wrinkled and does not lie level on the die. As the punches descend, they pierce the stock without leveling same, and as the blanks are afterward leveled, it is found that the pierced holes, being unevenly spaced, will not allow the blanks to interchange. By making the die, as shown in Fig. 78, with the stripper plate *M* fastened to the punch-holder and with a stiff coil spring at each corner, and so adjusted that the punches do not come

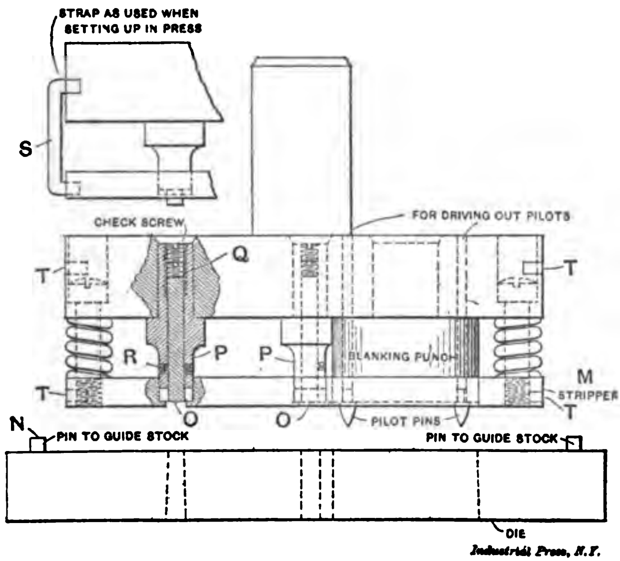


Fig. 78. Die with Stripper Attached to Punch to Flatten Stock

quite flush with the face of the stripper, the above-mentioned trouble is nearly eliminated. On the downward stroke of the press the stripper *M* presses the stock firmly against the die, holding it level while the punches perform their work. The stock is guided by means of a small pin *N* at each end of the die. The stripper should not fit the punches; for if the operator should make a miscut, or should a piece of scrap punching get under the stripper, it would cause it to tilt and bring disaster to the small punches.

Another valuable feature in this die is the manner in which the piercing punches *O O* are constructed. Ordinarily piercing punches are made solid, and if one breaks, it necessitates making a whole new punch or grinding the other punches down to the same length, greatly

shortening the life of the die. The punches shown at *O O* are designed to overcome this trouble. A holder *P* is made and left soft, into which the punch (or rod) *O* is inserted, being backed up by the screw *Q* and prevented from pulling out by means of the screws *R*. Then, should one of the punches "flake" off, that same punch can be ground and then forced out by means of the screw *Q* until it is at the same height as the others. This style of piercing punch greatly increases the life of a die. This die can be made either with or without the guide pins *E E* in Fig. 76. If made without the guide pins it is necessary to use the straps *S* to allow aligning the punches with the die when "setting up" in the press. The stripper is forced back and the straps inserted in the holes *T T*. After the die is "set up" and securely fastened, the straps are removed.

All presses in which double dies are used should be provided with a separator, which is a piece of sheet metal fastened underneath the press to separate the scrap punchings from the blanks. It is frequently

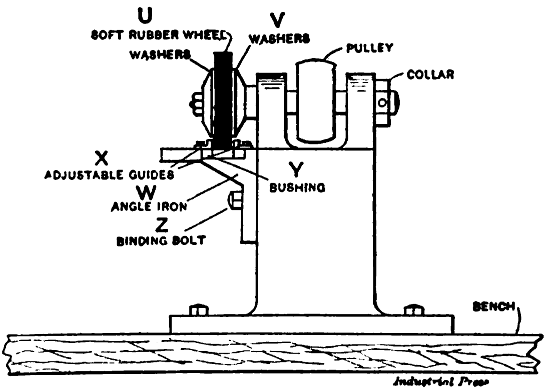


Fig. 79. Machine for Separating Blanks from Stock Strips

noticed that in factories where no separator is used, the cost of sorting the blanks from the scrap is in excess of the cost of blanking. A sub-press die leaves the blanks in a strip of stock. If the stock is over 0.02 inch thick, considerable trouble is experienced in removing the blanks. Fig. 79 shows a means whereby the blanks are forced from the strip without marring them. *U* represents a soft rubber wheel, which is supported on the sides nearly to the edge by the washers *V*. The angle iron *W* is provided with adjustable guides *X* and is recessed at *Y* to receive bushings having different sized holes. A bushing is inserted in the angle iron having a hole somewhat larger than the blanks to be forced out. The guides *X* are then adjusted to allow the strip to slide freely. The angle iron is then raised by loosening the bolt *Z* until sufficient pressure is brought on the rubber wheel. The wheel being power driven, all that is necessary is to place the end of a strip under the rubber wheel and it will roll the strip along, at the same time forcing out the blanks.

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NUMBER 9

DESIGNING AND CUTTING CAMS

THIRD EDITION—REVISED AND ENLARGED

CONTENTS

CHAPTER I

THE DRAFTING OF CAMS*

A cam is a device for converting circular into reciprocating motion. It generally consists of a disk having an irregular face that acts as driver of a follower in contact with it, or else of a groove cut in a flat or curved surface. Cams are very useful adjuncts to many forms of machines, as by their aid various complex and complicated movements may be obtained that were otherwise impossible. Their use is, however, attended with some objections of a character serious enough to warrant the substitution of some other method of arriving at a desired result when such other method is available. Among these objections may be mentioned the considerable amount of friction, producing wear, and the noisy action of cam movements. Despite these objections, cams have a wide use and are employed in many familiar machines.

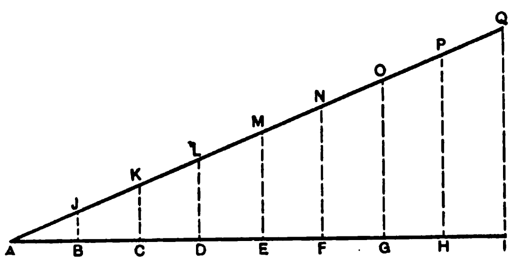


Fig. 1. Diagram graphically showing Motion Imparted to Follower by Cams in Figs. 2 and 3

Harvesters, printing presses, sewing machines, looms, and steam-valve mechanisms are a few of such machines to which cams contribute part of the action. The more complicated forms of automatic machinery, automatic screw machines, for instance, depend largely upon the aid of cams. The various machines used in the manufacture of shoes are also good examples of this class.

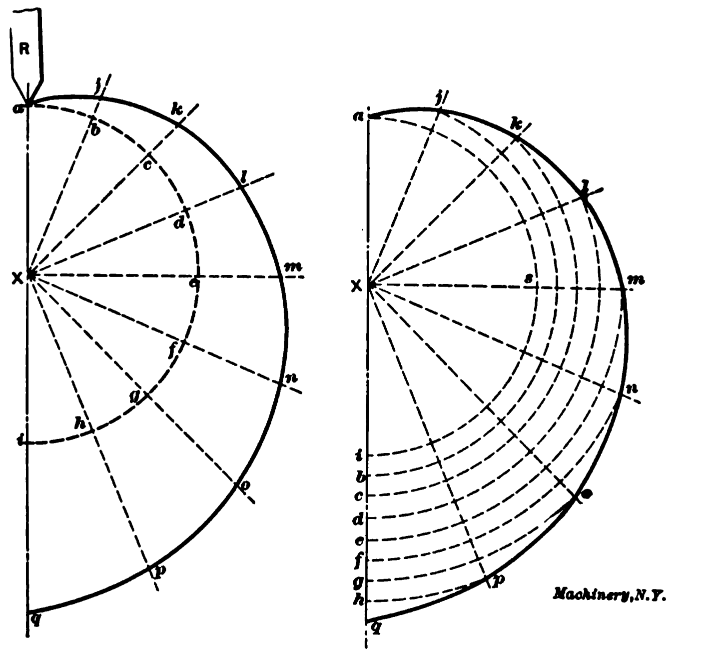
Laying Out a Cam for Uniform Reciprocating Motion

The knowledge of laying out cams is simply and easily acquired. The laying-out of a heart-shaped cam will serve as an illustration of the general method. This cam is used to convert circular motion into uniform reciprocating motion. Let it be required to lay out a cam that will move a follower with uniform velocity through a throw of 1½ inch. This action may be graphically shown by the aid of a diagram, Fig. 1. The action of but one-half the complete movement need be considered, as the return of the follower is along a curve similar to that occasioning the rise. Therefore, let *A I*, a line of indefinite length,

* MACHINERY, March, April and December 1897.

represent one-half a revolution of the cam. At *I* draw the perpendicular *I Q* equal to the extreme throw, in this case 1½ inch. As the rise of the follower is to be uniform, this action may be shown by a straight line connecting *A* and *Q*. Divide the line *A I* into any number of equal parts, say eight, and erect perpendiculars at the points of division. The point *E* will then represent one-quarter revolution of the cam, and the distance *E M* will represent the throw at that point. In the same way the distance *C K* represents the amount of throw at one-eighth revolution, the distance *G O*, the throw at three-eighths revolution, and so on for the other perpendiculars.

To lay out the cam curve, describe about *X*, Fig. 2, as center any semi-circle, *a e i*. Divide this semi-circle into the same number of



Figs. 2 and 3. Lay-out of Uniform Motion Cams

equal parts into which the line *A I* was divided. Connect these points of division with the center *X*, and extend the lines indefinitely beyond the semi-circle. On *X b*, make *b j* equal to *B J*, on *X c*, make *c k* equal to *C K*, and so on, extending each radius a distance equal to the corresponding perpendicular in Fig. 1. Then through the points *a*, *j*, *k*, *l*, etc., draw a smooth curve. This curve is one-half the required cam curve. By drawing a similar curve to the left of *a* the cam curve is completed. By rotating the cam about the center, *X*, the follower, *R*, would be forced to rise, with uniform velocity, through a distance of 1½ inch. During the second half of the revolution it would fall uniformly, by aid of gravity or a spring, to the initial point *a*.

Alternative Method of Laying Out Cam Curve

Another way of laying out the same cam curve is as follows: Draw any semi-circle, *a s i*, Fig. 3, and extend the diameter on one side a distance *i q* equal to the required throw. Divide *i q* into any number of equal parts, as at *b*, *c*, *d*, etc., and divide the semi-circle by the same number of radii equally distributed. With *X* as center and a radius

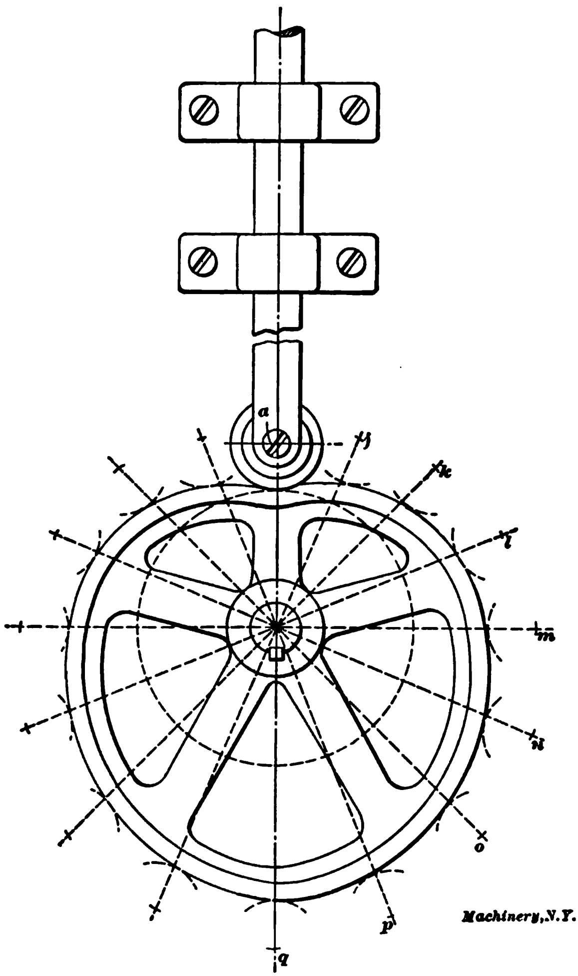


Fig. 4. Cam with Roller Follower

equal to *X b* describe an arc cutting *X j* at *j*. With the same center and radius equal to *X c* describe an arc cutting *X k* at *k*. Continue this process through the points *d*, *e*, *f*, etc., thus obtaining the points *l*, *m*, *n*, etc. The latter are points on the required curve.

The excessive friction of a pointed follower such as that shown at *R* necessitates the employment of a follower that will reduce the

amount of friction to a minimum. A small roller meets this requirement. If a roller is employed as a follower the problem of laying out the cam curve becomes modified. A roller traveling along the curves shown in Figs. 2 and 3 would not impart to the follower-rod the desired uniform rise and fall. The variation would be but slight, yet sufficient to merit consideration where accuracy is desired.

Cams with Roller Followers

Fig. 4 represents a heart-shaped cam of the same dimensions as in Figs. 2 and 3, but with a roller follower. It is the path of the center

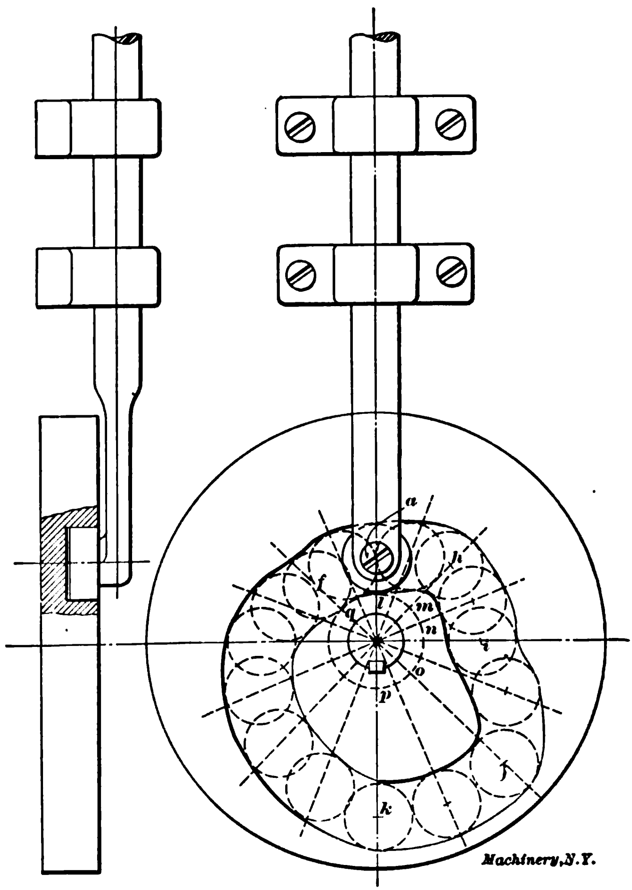


Fig. 5. Positive Action Cam for Variable Motion of Follower

of this roller that requires the first consideration, as the position of this center regulates the throw. Therefore, the position of the center of the roller at various intervals in the rotation of the cam must be determined. This may be done by adding to each of the distances *J B*, *K C*, *L D*, etc., in Fig. 1, the radius of the roller, and thus obtaining

the points j, k, l, etc. With these points as centers and with radii equal to that of the roller, describe arcs. A curve drawn tangent to these arcs is the required cam curve.

This cam depends upon the action of gravity, or a spring, to keep the follower in contact with the driver. It can be made positive in action by the use of two followers placed at the extremities of the diameter of the cam, or by drawing curves tangent to both the top and bottom of the follower roller in its various positions, and the two curves taken as the boundaries of a groove cut into the metal. A familiar application of the use of a heart-shaped cam may be found in the bobbin-winder of the domestic sewing machine. The thread is fed to and fro at a uniform rate, the follower of the cam acting as a guide for the thread. The action is made positive by the employment of two follower rollers.

Positive Action Cam for Variable Motion of Follower

The latter method of laying out a positive motion cam referred to above is more clearly shown in Fig. 5. A variable motion is here substituted for the regular motion of the heart-shaped cam. Let it be required to lay out a positive motion cam that shall impart to the

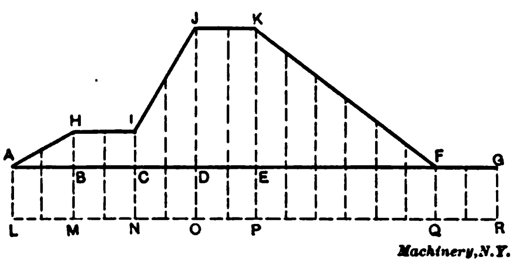


Fig. 6. Diagram of Motion Imparted to Follower by Cam in Fig. 5

follower the following action: A uniform rise of ¼ inch during the first eighth of a revolution; no action during the next eighth; a uniform rise of ¾ inch during the third eighth; no action during the fourth eighth; a uniform fall of 1 inch during the next three-eighths of the revolution; and no action during the last eighth. The action is graphically shown in Fig. 6. Let AG represent one complete revolution of the cam; B, the first eighth; C, the second; D, the third; E, the fourth; and F, the seventh. The problem calls for a uniform rise of ¼ inch during the first eighth. Therefore, from B draw the perpendicular BH, ¼ inch in length, and join A and H. As there is to be no action during the second eighth, draw HI parallel to BC; that is, the follower will be the same distance from AG at I that it was at H, and therefore the follower will not have been acted upon. During the next eighth revolution the follower is required to move ¾ inch. As it has already moved ¼ inch, the sum of these two distances is the length DJ. As this rise is to be uniform, a straight line is drawn joining I and J. No action during the fourth eighth is shown by drawing JK parallel to DE. A uniform fall of 1 inch during the next

three-eighths of the revolution is shown by joining K and F, and the period of rest during the last eighth revolution is shown at FG. The line AL is equal to the radius of the roller, and by drawing the line LR parallel to AG, the distance of the center of the roller from the base circle may be taken directly for any radius of the cam.

To lay out the cam from the diagram, draw any base circle $l n p$, Fig. 5, and divide it into the same number of equal parts into which the line AG is divided, *viz.*, sixteen. Through these points of division

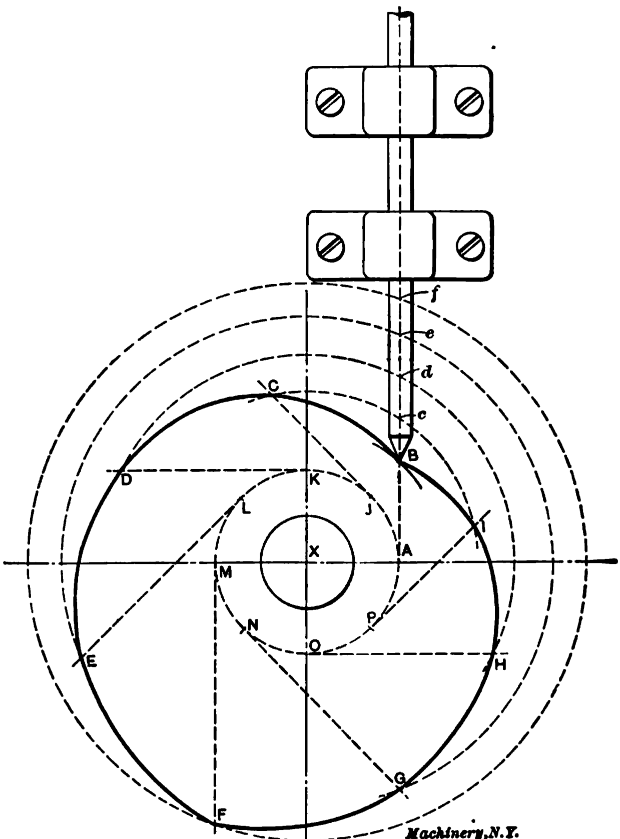


Fig. 7. Cam with Follower having Line of Action Eccentric with Cam Axis

draw radii and extend them indefinitely. Upon these radii take $la = LA$, $mh = MH$, $ni = NI$, etc., thus determining the positions of the center of the roller at the various intervals. Sketch in the outline of the roller in its different positions, and draw curves tangent to these outlines.

Line of Action of Follower Eccentric with Cam Axis

In the cams previously considered, the line of action of the follower passes through the center of the cam-shaft. When the line of action

of the follower passes to either side of the center of the cam-shaft, as in Fig. 7, a different method of laying out the cam curve becomes necessary. Assume that the requirements and conditions are the same as in Fig. 2, excepting that the line of action of the follower shall be one inch to the right of the center of the cam-shaft. Draw the indefinite line *X A* passing through the center of the cam-shaft. One inch to the right of *X* draw the line of action *A f*, of the follower, perpendicular to *X A*. Let *B* be the lowest position that the follower is to assume, and let *f* be the highest. Divide the throw, *B f*, into any number of equal parts, as at *c*, *d* and *e*. Through *A* describe a circle with *X* as center. Divide this circle into twice the number of equal parts into which *B f* is divided. From each of these points *J*, *K*, *L*, etc., draw tangents to the circle. Then, with *X* as center, describe arcs through *c*, *d*, *e*, and *f*. Where the arc *c* cuts the tangents from points

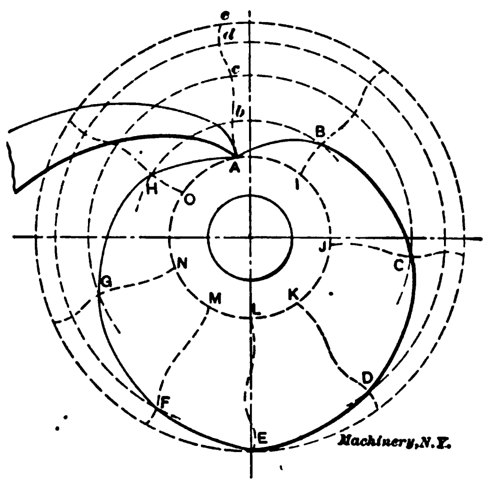


Fig. 8. Cam and Follower both having Variable Motion

J and *P*, as at *C* and *I*, are points on the desired curve. Where the arc through *d* cuts the tangents from *K* and *O*, as at *D* and *H*, are also points on the curve. The points *E*, *F*, and *G* are obtained in a like manner.

Cams with Pivoted Followers

The problem in Fig. 2 may be further modified by having the follower pivoted instead of acting in a straight line. In this case, the line of action becomes the arc of a circle. Problems of this nature may be solved by substituting for the straight line of action shown at *i q*, Fig. 3, an arc which shall represent the path of the follower. This arc of action takes the place of all the various radii in Fig. 3, and the points *b*, *c*, *d*, etc., serve as a series of initial points from which to swing concentric arcs to intersect the various positions of the arc of action of the follower. The method is analogous to that in Fig. 3. In Fig. 29 this method is applied to a cam of un-uniform motion.

Cams and Followers both having Variable Motion

The rotation of the driver has thus far been considered as uniform, and the action of the follower either uniform or irregular. A case will now be considered wherein both the action of the driver and that of the follower is irregular. In Fig. 8, let the unequal divisions into which the base circle *A J L* is divided by the points *A*, *I*, *J*, etc., represent spaces traversed by the driver in equal periods of time. That is, if it takes the driver one second to rotate through the arc *A I*, it will take the same time to rotate through the larger arc *I J* or the smaller arc *L M*. Again, let *A e* represent the irregular path of the follower and the points *b*, *c*, *d*, and *e* its position at certain equal intervals of time, say one second. The number of divisions made in the path

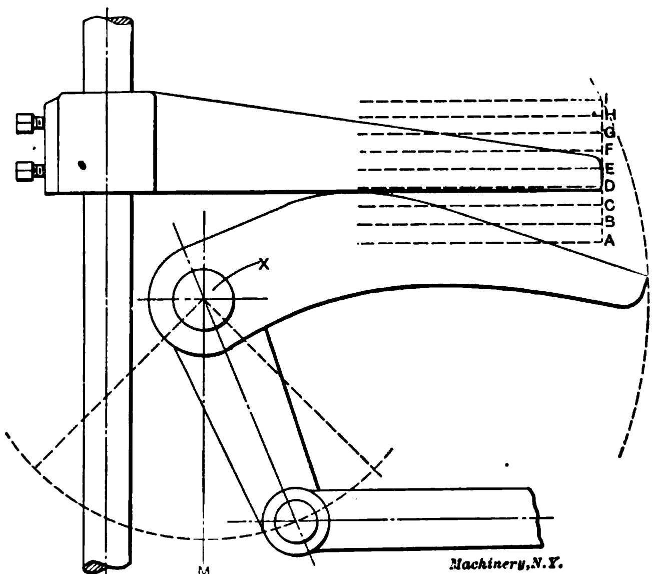


Fig. 9. Cam with "Flat-footed" Follower

of the follower should correspond with the number of divisions into which one revolution of the driver is divided. The points *B*, *C*, *D*, etc., of the cam curve may be found by the method of intersections explained in Fig. 3. This problem is of a general nature and is universally applicable to problems involving a disk driver and a follower other than a flat-footed one.

The "Flat-footed" Follower

A familiar example of a flat-footed follower is afforded by the toe-and-lift mechanism used to actuate the engine valves of side-wheel steamers. The "lift" or "wiper" is pivoted upon a rock-shaft which is caused to oscillate by an eccentric placed upon the paddle-wheel shaft. In Fig. 9, let the arc through which the rock-shaft swings equal 90 degrees—45 degrees on either side of the vertical—and let

the "toe" rise and fall with uniform motion through 1½ inch. It is required to design the upper face of the lift to give the desired throw.

Divide the throw, *A I*, into any number of equal parts, say eight, and locate the center of the rock-shaft, as *X*. Upon a piece of tracing paper draw a quadrant, *x h k*, Fig. 10, *x k* being equal to one-half the throw of the eccentric, say 3 inches. Draw *x l* at 45 degrees to *x h*, and *k l* at right angles to *x k*. Through the point of intersection, *l*, and with *x* as center, describe the arc *l m*. The arc *k h* then represents a quarter revolution of the eccentric, and the arc *l m* the corresponding angular movement of the rock-shaft crank. Divide the arc *k h* into the same number of equal parts into which the throw of the toe was divided, *viz.*, eight. Through these points of division draw lines parallel to *x m*, intersecting the arc *m l* in the points *n*, *o*, *p*, etc. From these points draw radial lines. Now, while the eccentric is moving through a quarter revolution with a uniform motion, as shown

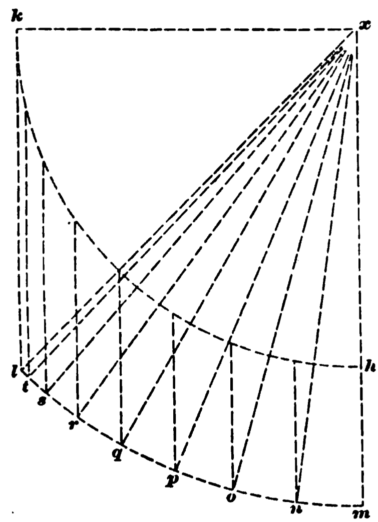


Fig. 10. Lay-out for Cam with "Flat-footed" Follower

by the equal division of the arc *h k*, the center line of the rock-shaft crank will assume the corresponding positions shown by the radial lines.

Place the tracing in Fig. 10 upon Fig. 9 so that *X* and *x* coincide, and the line *x m* falls upon the line *X M*. Then draw upon the tracing-paper the position of the line *A*. Rotate the tracing-paper about *X* until *x n* coincides with *X M* and draw the position of the line *B*. Again rotate about *X* until *x o* coincides with *X M* and draw the position of the line *C*. Continue this process until the positions of the lines *D*, *E*, *F*, etc., are located. A curve drawn tangent to the lines thus obtained is the required cam curve. This latter procedure is not shown in the cuts. The use of tracing-paper for laying out cam curves, as here

exemplified, is applicable to the laying-out of a variety of such curves. The tracing may be made to assume different positions of either the driver or follower, and their relation shown at any desired interval during their action.

In work dealing with cam curves there are some factors of a practical nature that must be considered, one of which may be here stated, as applying directly to the problem of the toe-and-lift. This factor is the easement of cam action to prevent jerking. The action as drawn in Fig. 9 has too abrupt a beginning and ending, and should be modified by an easement curve at both these points of action. In any action that tends to jerkiness, a smoother motion may be obtained by slightly modifying the curve at the offending point.

Cams with Double Contact

In the drawings of cams thus far shown, there has been but one point of contact between the driver and follower. Positive motion is often obtained by having two points of contact. Cams having two such

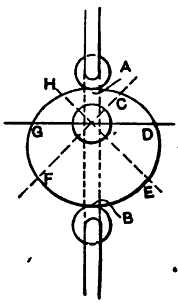


Fig. 11

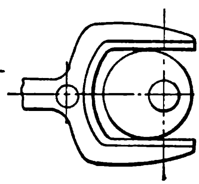


Fig. 12

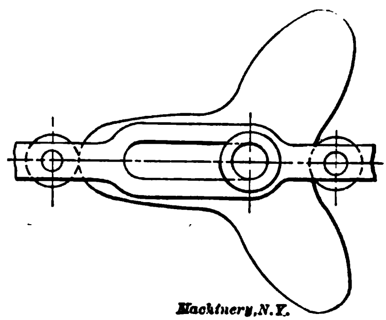


Fig. 13

points of contact are subject to certain limitations. For instance, in Fig. 11, if *A* and *B* are two points of contact of the follower, and are a constant distance apart, and the curve *A D B* be any assumed curve of one-half revolution of the cam, the curve of the remaining half revolution is limited to a curve complementary to *A D B*. That is, the distances *C F*, *D G*, and *E H* must equal the constant *A B*.

If it is desired to have an independent movement throughout the entire revolution of the cam it will be necessary to have two cams placed one upon the other, one point of contact of the follower bearing upon the second cam. In this case, having assumed any curve for one of the cams, the other cam must be made complementary to the first, the constant distance apart of the points of contact forming the basis for the calculation. Forms of double contact cams are shown in Figs. 12 and 13. Fig. 12 is a rocker cam, and Fig. 13 is a tri-lobe cam giving three reciprocating motions to the follower for each revolution of the driver.

Cylindrical Cams

Fig. 14 illustrates a method for laying out cylindrical cams. Let *g d a* be the plan, and *H a'* the development of the cylinder shown

in elevation at *K A*. Divide the plan into any number of equal parts as at *a*, *b*, *c*, etc., and project these points of division upon the front elevation of the cylinder as the elements *A*, *B*, *C*, etc. On the developed surface these elements appear as *a′*, *b′*, *c′*, etc. Upon the development, lay out the desired action, as in Figs. 1 and 6, avoiding or easing all sharp corners. Suppose *m l p* to be such an action. This curve will then represent the path of the center of the follower. Let *L* indicate the center of the follower. Then, as the cylinder is rotated about its axis, the point *L* moves to and fro a distance *L I*, and with an irregular motion dependent on the form of the curve *m l p*. The projection of this curve upon the elevation of the cylinder is shown at *L m*.

The form of the roller-follower may be either cylindrical or conical; the question of the shape of the follower has been treated more completely in Chapter V. In laying out the cam practically, the outline of the groove may be drawn by the method shown in Fig. 5, that is,

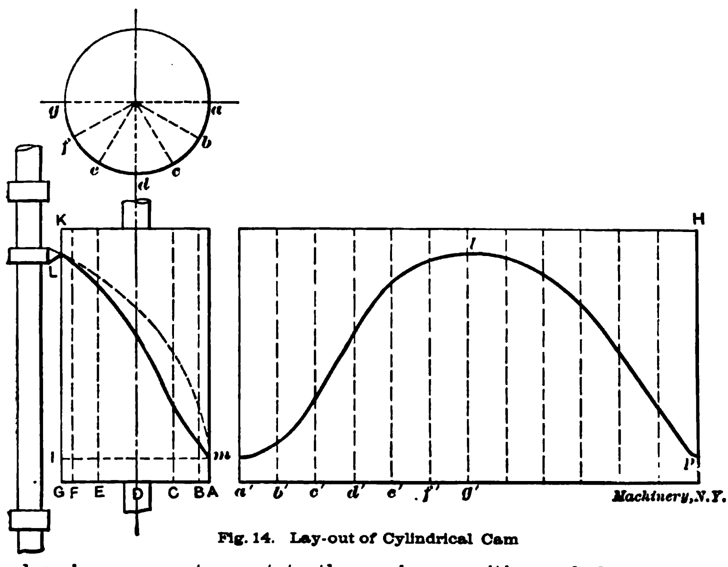


Fig. 14. Lay-out of Cylindrical Cam

by drawing curves tangent to the various positions of the roller, and then, by winding the drawing about the metal cylinder blank, any number of points of the groove may be located with a prick-punch; or, the drawing may be made directly upon the surface of the cylinder.

The method for laying out a conical cam is similar in principle to that for laying out a cylindrical cam, and is easily deduced from the latter.

Laying Out a Cam for Shifting Planer Belts

The following problem in machine design is one of a series given to the students in mechanical engineering at Cornell University. It furnishes a good example of the method of reasoning applied to practical problems in mechanics, and is also an interesting problem in quick-return motions. The problem calls for the designing of a device

for automatically shifting the belts of a planer. The driving shaft has a fixed pulley of wide face carrying two belts. The driven shaft has two sets of a loose and a fixed pulley. One set, smaller than the other, is driven by a crossed belt, and its shaft therefore rotates in a direction opposite to that of the driving shaft. The larger fixed pulley drives the planer while the tool is cutting, and the smaller fixed

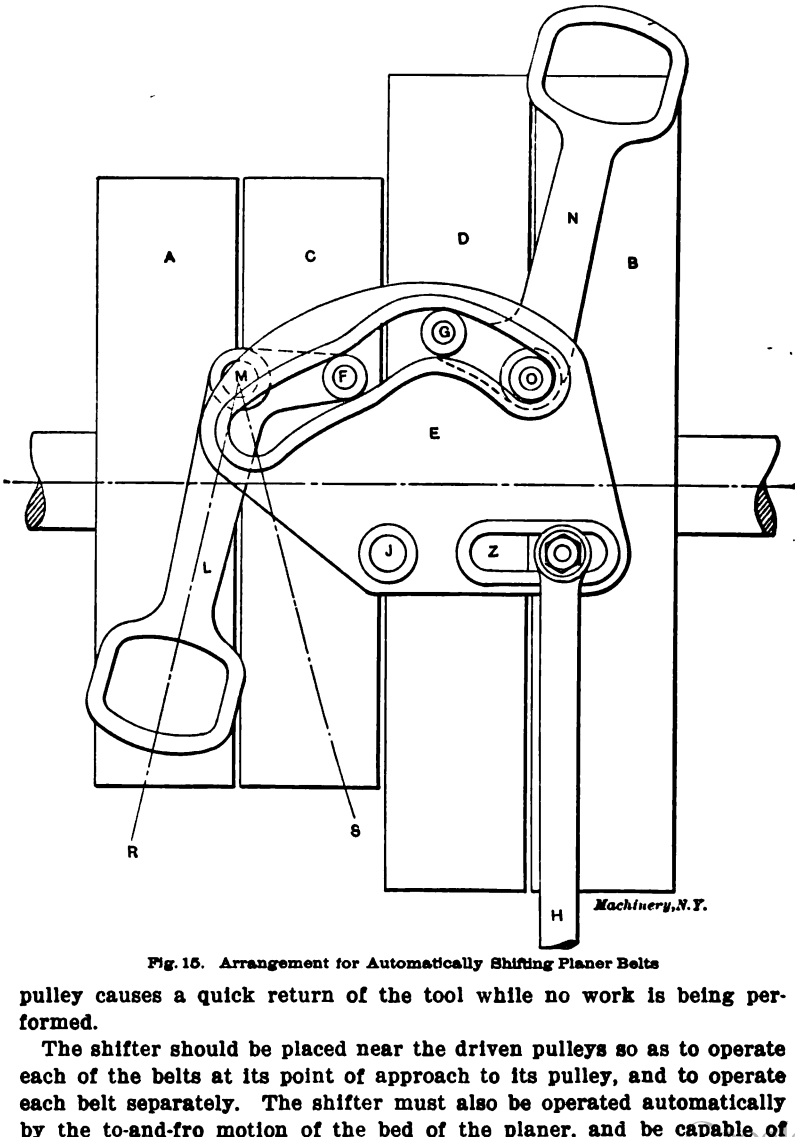


Fig. 15. Arrangement for Automatically Shifting Planer Belts

pulley causes a quick return of the tool while no work is being performed.

The shifter should be placed near the driven pulleys so as to operate each of the belts at its point of approach to its pulley, and to operate each belt separately. The shifter must also be operated automatically by the to-and-fro motion of the bed of the planer, and be capable of

adjustment to allow for the variation of the momentum of the machine under different loads.

In Fig. 15, *A* and *B* are the two loose pulleys of the driven shaft, and *C* and *D* the fixed pulleys. *E* is a grooved cam rotating about *J*, and having two roller followers *F* and *G*. *H* is a link driven to and fro by a tripping device attached to the planer bed. *L*, the shifter-arm for the smaller pulleys, is a crank rotated by the follower *F* about *M* as a center. In a similar way, the crank *N* rotates about *O*. The pivots *J*, *M* and *O* are carried on a plate made fast to the planer and not shown in the drawing. The portions of the cam to the left of *F* and to the right of *G* are arcs of circles with *J* as a center, and there-

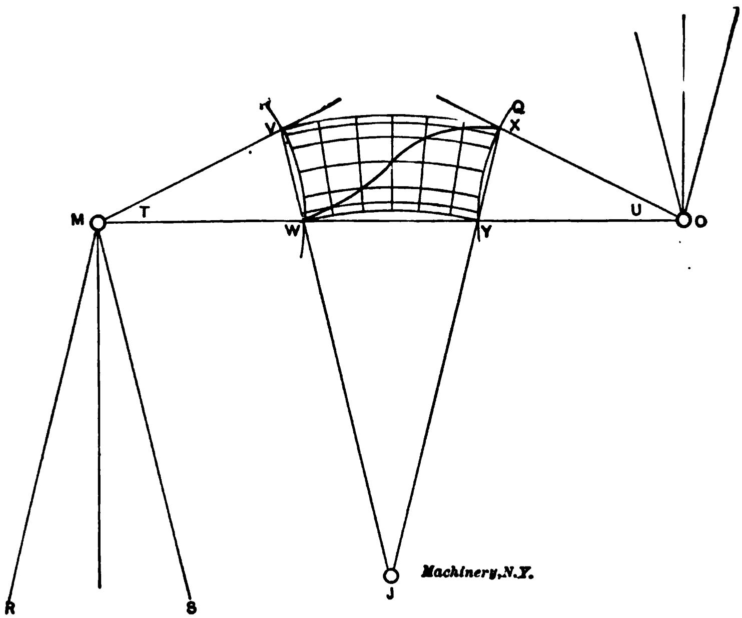


Fig. 16. Lay-out of Cam Curve for Cam in Fig. 15

fore, while either of the followers is traveling through these arcs there will be no movement of the shifter-arms. The throw of either of the arms is occasioned by its follower traversing the irregular path between *F* and *G*.

Imagine the link *H* drawn downwards. The cam then rotates towards the right about the center *J*. The follower *F* is held fixed in its position by the arc of the cam to its left, and therefore the shifter-arm *L* remains stationary. The path of the follower, *G*, however, is through the irregular part of the cam between *F* and *G*, which causes it to rotate about *O* as a center, thereby shifting the arm *N* from the loose pulley *B* to the fixed one *D*. If the link *H* is operated in the reverse direction to that imagined above, the shifter-arm *L* will then become the active member, and the shifter-arm *N* will remain inoperative.

A method for determining the irregular path of the centers of the followers *F* and *G* is shown in Fig. 16. First locate the points *M* and *O* from Fig. 15, and draw the circular arcs *P* and *Q*, the paths of the centers of the followers *F* and *G*. Then draw *R* and *S*, the extreme positions of the center line of the shifter-arm *L*. Make angles *T* and *U* equal to the angle formed by the lines *R* and *S*. Divide the line through *VW* into six parts proportional to 1, 3, 5, 5, 3, 1, and through the points of division draw arcs with *J* as a center. Divide *VX* into six equal parts, through which draw radial lines. The successive intersections of the circular arcs and the radial lines determine the paths of the followers *F* and *G*, as *WX*. Lines drawn tangent to successive positions of a follower along the line *WX* will be the outline of the cam-slot at its irregular part.

The slot *Z*, Fig. 15, permits adjustment of the link as called for in the conditions of the problem. The center of the opening for the belt in the shifter-arm *L* is placed nearer to the center line of the shaft to allow for the angularity of the cross belt.

Laying Out an Intermittent Motion Cam with Pivoted Follower*

The cam to be laid out is shown in Fig. 17. It turns toward the left and moves a 1-inch roller *A* which controls the lever *B* swinging on the stud *C*. The cam is to be keyed to a shaft, together with several other cams, in all of which the keyway is at the beginning of the cycle. The requirements which follow are selected to illustrate as simply as may be the method employed. The head of the lever *B*, which is 12⅝ inches long, is to remain at rest until the cam has turned 150 degrees from the zero point or beginning of the cycle; it is then to advance 1½ inch in 43 degrees; then it will dwell for 35 degrees more, and, finally, retreat 1½ inch in 92 degrees, after which it will dwell for the remainder of the cycle. In Fig. 17 it is seen that the roller *A* is located at one-third of the distance from the pivot of the lever to its head. Hence a movement of one-half inch is required of the roller in the cam to move the lever head 1½ inch.

We will now begin the lay-out. Draw first the circumference of the cam; its diameter we will make 10 inches. With the keyway on the vertical diameter, draw a line through its center. With this line as zero, divide the circumference into 30-degree sections, as shown, and number them. Now draw the circle *D* with a radius of 4 3/16 inches, to show the extreme outer position of the center of the roller, and the circle *E* with a radius of 3 11/16 inches, to show the extreme inner position of the center of the roller. Next, with the center of the cam as its center, draw the circle *F*, so that it will pass through the center of the stud *C*. Beginning with the center of the stud *C* as zero, divide this circle into sections and number them, as shown, for each 60 degrees. Such further sub-divisions as may be needed later may be made when required.

Proceed now with care to place the needle of a pair of good compasses in the center of the roller *A*, and adjust them so that the pencil

* Herbert C. Barnes, MACHINERY, October, 1908.

point will pass through the center of the stud *C*. We will call this radius *R*. Now having in mind the requirements stated above, one being that the cam should turn 150 degrees from its zero before the roller moves, place the end of the compasses at 150 degrees on the circle *D*. Holding the needle here, with the radius *R* draw an arc intersecting the stud circle *F* at the point *G*. It is seen that the point of intersection is at 60 degrees on the circle *F*. Now place the needle point 43 degrees further along on the stud circle, or at 103 degrees, and with the radius *R* draw an arc intersecting the circle *E* at the point *H*. The point *H* marks the halt of the advance of the roller,

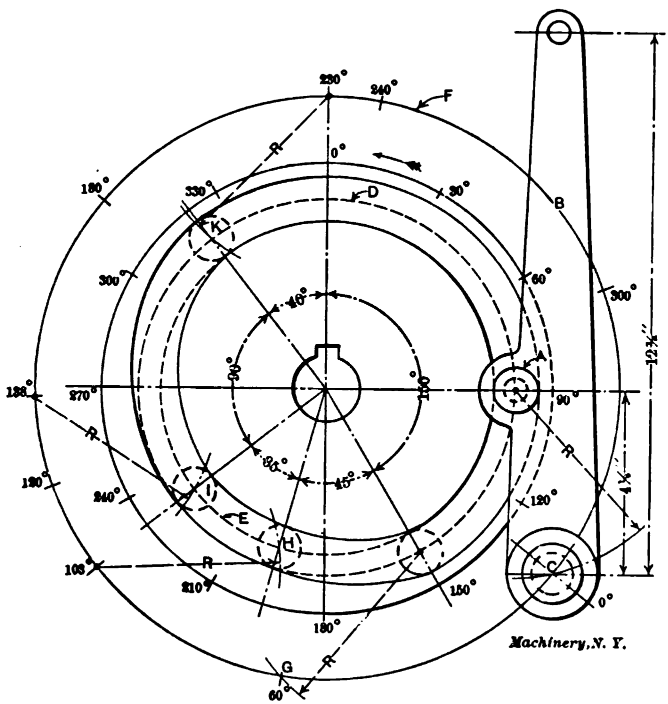


Fig. 17. Lay-out of Intermittent Motion Cam with Pivoted Follower

and the beginning of its dwell. Now move the needle 35 degrees further along the stud circle to 138 degrees, and with the radius *R* draw another arc intersecting the circle *E* at the point *I*. This point marks the end of the dwell and the beginning of the retreat. Now move the needle 92 degrees further along the stud circle to 230 degrees and with the radius *R* draw an arc intersecting the circle *D* at the point *K*. This point marks the end of the retreat and the beginning of the dwell for the remainder of the cycle.

The points *H*, *I* and *K* being marked, draw radii through them extending to the circumference of the cam circle. Knowing that the roller begins to advance at 150 degrees on the cam, the advance is seen to

continue for 45 degrees. The roller then dwells for 35 degrees and retreats in 90 degrees, after which it dwells until the next advance begins. It is proper that these figures do not agree with the figures for the lever movement stated above. Barring possible slight errors in the lay-out, they are correct for the cam.

The radius of the inner wall of the raceway or groove is, of course, ½ inch less than that of the path of the cam center. Hence the radius of the inner wall of the outer dwell is 3 11/16 inches, and that of the inner dwell is 3 3/16 inches. This inner wall is the counterpart of the master cam which will be used for cutting the cam groove.

CHAPTER II

CAM CURVES*

When the curve of a cam is not determined by a given definite motion of the follower, and the condition presented to the designer is simply to make the follower move through a given distance during a given angle of motion of the cam-shaft, the ease and silence with which the cam works depends upon the character of curve used in laying out the advance and return. The uniform motion curve, the simplest of all curves to lay out, is a hard-working curve, and one that cannot be run at any great speed without a perceptible shock at the beginning and end of the stroke.

Uniform Motion Curve

The uniform motion curve would be represented in a diagram by the diagonal of the rectangle of which the base represents the angle of motion, and the altitude, the stroke of the cam, as shown by the full lines in Fig. 18. However, should the nature of the design demand a uniform motion for a given part of the revolution of the cam-shaft, the shock at beginning and end of stroke may be modified by increasing both the angle of motion and the stroke, and, in the diagram, filling in arcs of circles as shown by the dotted lines in Fig. 18. The amount of curvature at the ends of the stroke is dependent upon the amount it is possible to increase the angle of motion, and the centers of the arcs are determined by drawing perpendiculars to *X Y* as shown in Fig. 18. It will be noticed that the uniform motion has been maintained for the original angle, the modifications at the ends causing the increase of angle of motion and of stroke, the rectangle formed by these two being shown by dotted lines. Even with these modifications the cam is still apt to work hard, especially if the angle of motion is small.

* MACHINERY, April, 1907; July, 1907, and February, 1908.

Harmonic Motion Curve

The crank or harmonic motion curve works much more easily than the uniform curve, and a cam laid out with this motion may be run at a high speed without much shock or noise. To draw a diagram of this curve, draw a semi-circle having a diameter equal to the stroke of the cam, and divide this semi-circle and the line representing the angle of motion into the same number of equal parts. The intersection of lines drawn from these divisions will give points on the curve. Fig. 19 shows the harmonic curve and the manner in which it is obtained.

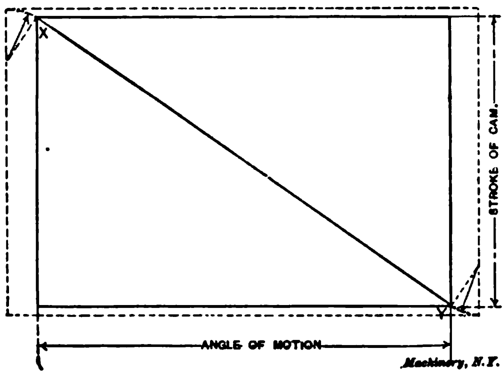


Fig. 18. Uniform Motion Curve

Gravity Curve

Probably the easiest working cam curve is the one known as the gravity curve. This curve has a constant acceleration or retardation bearing the same ratio to the speed as the acceleration or retardation produced by gravity; hence its name. A body falling from rest will pass through about sixteen feet in one second (more accurately 16.08 feet). During the next second the body will increase its velocity by about thirty-two feet making the distance covered during the second

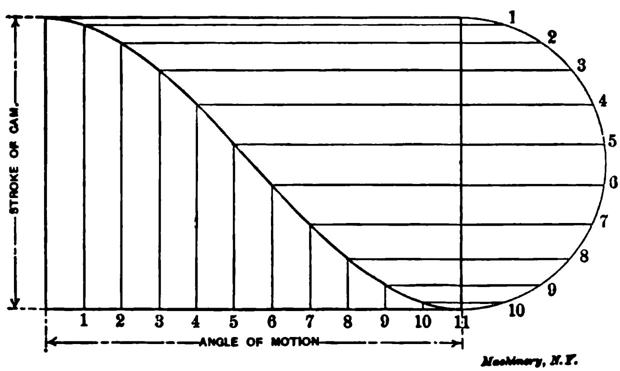


Fig. 19. Crank or Harmonic Motion Curve

second forty-eight feet; during each succeeding second the body will gain in velocity thirty-two feet. Using sixteen feet as a unit of measurement, it will be seen that a body would travel through units 1, 3, 5, 7, 9, etc., during successive seconds or units of time. To apply this motion to the cam curve, we might divide the angle of motion into a given number of equal parts and, using the units given above, we may increase the velocity to a given maximum and then, retarding with the same ratio, bring the follower again to rest at the other end of the stroke. In the diagram, Fig. 20, the line representing the angle of motion is divided into eleven equal parts which necessitates

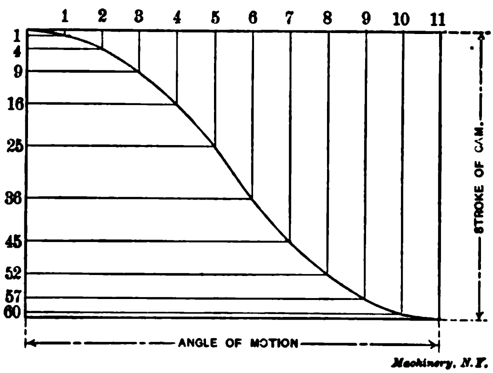


Fig. 20. Gravity Motion Curve

eleven divisions on the line representing the stroke of the cam. If the motion for the first part of the stroke is to have a constant acceleration, as referred to above, the distance traversed by the follower during the first part of the angle of motion would be one unit; in the second part, three units; in the third part, five units, and so on until the maximum velocity had been reached which would be during the

Number of period.	Distance traversed by follower during one period.	Total distance traversed since beginning of motion.
1	1	1
2	3	4
3	5	9
4	7	16
5	9	25
6	11	36
7	9	45
8	7	52
9	5	57
10	3	60
11	1	61

sixth part of the angle of motion when the follower would travel through eleven units of motion. At this point the motion would begin to be retarded by a constant deduction which would cause the follower to move through nine units during the seventh interval of time, seven units during the eighth, five units during the ninth, three units during

the tenth, and one unit during the eleventh and last interval. The sum of these units is sixty-one, which will necessitate dividing the line representing the stroke of the cam into sixty-one equal parts of which the first, fourth, ninth, sixteenth, twenty-fifth, thirty-sixth, forty-fifth, fifty-second, fifty-seventh, sixtieth, and sixty-first will be used for determining points on the curve. The combination of the table given and the diagram shown in Fig. 20 will show how the gravity curve may be drawn.

Approximation of Gravity Curve

A very close and satisfactory approximation for the gravity curve, and one that entails less work than the theoretical, is shown in Fig. 21. The method of drawing is similar to the one used for the harmonic motion, excepting that an ellipse takes the place of the semi-circle. It can be seen very readily that the ratio of the major and minor axes will determine the character of the cam curve. To obtain a curve that

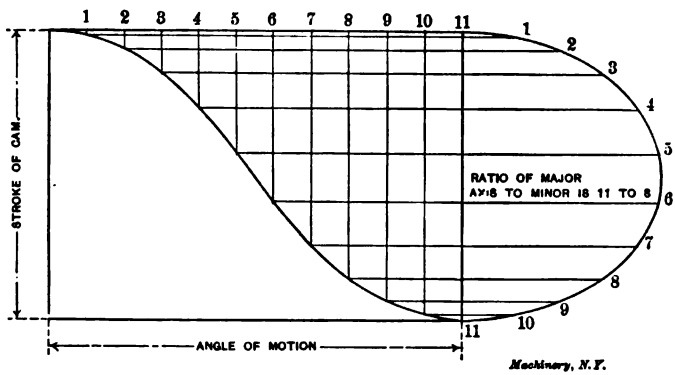


Fig. 21. Approximate Gravity Curve

will approximate the gravity curve, the line representing the stroke of the cam should be used as the minor axis and the ratio of major axis to minor axis should be 1⅜ to 1 or 11 to 8. Dividing the semi-ellipse and line of angle of motion into the same number of equal parts, and projecting, we obtain points on the curve. Fig. 22 is given so that a comparison may be made of the three motions given above when applied to the same cam.

Laying Out Cams for Rapid Motions

As already mentioned in Chapter I, we may consider a cam mechanism as being made up of two elements. As generally constructed, one element is a revolving plate cylinder, cone or sphere, and the other element is a bar or a roller which has some form of reciprocating motion. The revolving piece is usually made the driver, although the mechanism may be made to work in the reverse order. The shape of a cam will depend upon the kind of motion that the follower is required to have. The motion of cams that are used for driving parts

of machinery, may be, as we have already seen, one of three kinds, *viz.*:

1. *Uniform motion*, in which the follower is made to pass over equal spaces in equal intervals of time.

2. *Simple harmonic motion*, in which the follower is accelerated from rest to a maximum velocity and then retarded again to a state of rest, following the harmonic cycle.

3. *Uniformly accelerated motion*, in which the follower is accelerated from rest to a maximum velocity and then retarded again to a state of rest, the acceleration being uniform, as, 1 inch per second, 2 inches per second, etc.

To this we may add a fourth kind frequently met with:

4. *Intermittent motion*, periods of motion being interrupted by periods of rest.

In slow-moving machinery it may not be important whether the follower moves with uniform, simple harmonic, or uniformly accelerated motion, but in machines where the cams have a high rotative speed, and the follower a reciprocating motion, as in the case of sewing machines and in some textile machinery, a uniform rate of motion will

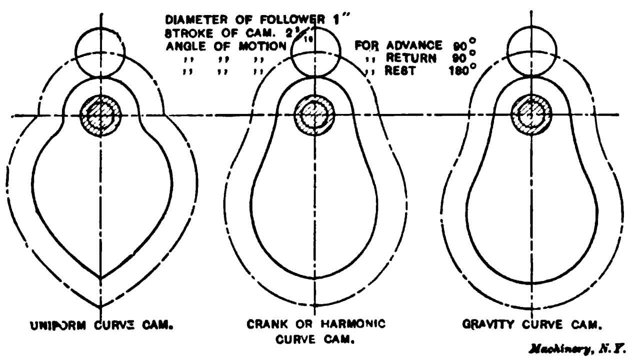


Fig. 22. Comparison between the Different Cam Constructions

be unsatisfactory or impossible. The reason for this is that the follower is impelled from rest to its maximum velocity instantly, and also brought to rest from a maximum velocity instantly. This gives it a sudden jerk at each end of the motion, which is very trying to a machine when the reversals take place rapidly. Cams for high rotative speeds, where the follower has a reciprocating motion, should, therefore, be so designed that the follower will start gradually, attain its maximum speed near the middle of its path, and then gradually come to rest. In other words, the follower should have a uniformly accelerated motion during the first half of its movement, and a uniformly retarded motion during the last half.

In uniformly accelerated motion $S = \frac{1}{2}Pt^2$, where S = the distance passed over, P = the acceleration, and t = the time. This is the same as saying that the distance which the body has passed over at the end of any number of units of time varies as the square of the number of such units. For example, if a body has a uniform acceleration of 2

inches per second, $S = \frac{1}{2} \times (2) \times (1)^2 = 1$ for the first second; $S = \frac{1}{2} \times (2) \times (2)^2 = 4$ for the next second; and so on. This is, as said before, also the law of falling bodies whose motion is not resisted by the air or other medium. Uniformly retarded motion obeys the same law. If time intervals of such a motion be plotted as abscissas and the corresponding space intervals as ordinates, with reference to coordinate axes, the resulting curve will be a parabola, and this is the curve that should be used for the outline of cams that are designed for high rotative speeds.

Uniform Motion Cylinder Cam

The cams shown in the following cuts do not necessarily represent any existing forms; they simply illustrate how the principle may be applied to certain shapes of cams and paths of followers. In Fig. 23, lay out on a sheet of paper $A\,B\,D\,C$ a line constructed as follows: Bisect $C\,D$ at M and divide $C\,M$ into any convenient number of parts, say

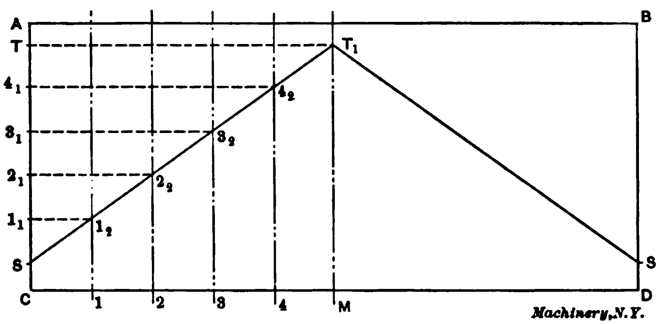


Fig. 23. Development of Uniform Motion Curve

five. Lay off on $C\,A$ any distance $S\,T$, and divide $S\,T$ into the same number of parts as there are in $C\,M$. Through the points 1, 2, 3, etc., on $C\,M$, erect perpendiculars to $C\,M$, and through the points 1_1, 2_1, 3_1, etc., on $C\,A$, draw parallels to $C\,M$ intersecting the perpendiculars at points 1_2, 2_2, 3_2, etc. A line $S\,T_1$ drawn through these intersections will be straight. The line $T_1\,S_1$ can be found in the same way. Now if the sheet of paper $A\,B\,D\,C$ be wrapped around the outside of a cylinder whose circumference is equal to the distance $C\,D$, the line $S\,T_1$ will take the position $S\,T$, Fig. 24, and the line $T_1\,S_1$ will form a similar curve on the reverse side of the cylinder. If this curve be made the center line of a groove, as the cylinder revolves on its axis, the groove will drive a follower up and down, parallel to the elements of the cylinder, with a uniform speed. The follower will start and stop at either end of its motion with a sudden jerk.

Uniformly Accelerated Motion Cylinder Cam

In Fig. 26 let $A\,B\,D\,C$ represent the paper as before. Bisect $C\,M$ at 3, and $S\,T$ at 9. Divide $C3$ and $3M$ into any convenient number of parts, say three; then divide $S9$ and $9T$ into the square of three parts, or 9, as shown. Erect perpendiculars to $C\,M$ at the points 1, 2, 3, etc., and draw parallels to $C\,M$ through the points 1, 4, 9, 4_1, and 1_1. Through

the points S and T_1 and the intersections 1_1, 2_1, 3_1, $2'_1$ and $1'_1$, draw a smooth curve. This line will be a parabolic curve, reversing at 3_1. The curve $T_1 S_1$ is constructed in the same way. Now wrap the sheet of paper $A B D C$ around a cylinder whose circumference is equal to $C D$. The curve will take the position $S T_1$, Fig. 27, and the curve $T_1 S_1$ will take a similar position on the reverse side of the cylinder. A groove made with these curves as center lines will drive a follower P up and down through the distance K, as the cylinder is rotated on its axis. The follower will start gradually at S, attain its maximum velocity, and then come gradually to rest again at T_1, the motion being

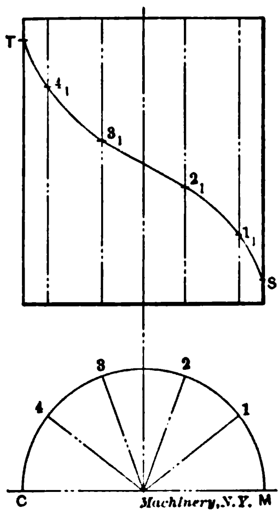


Fig. 24. Uniform Motion Curve scribed on Cylindrical Surface

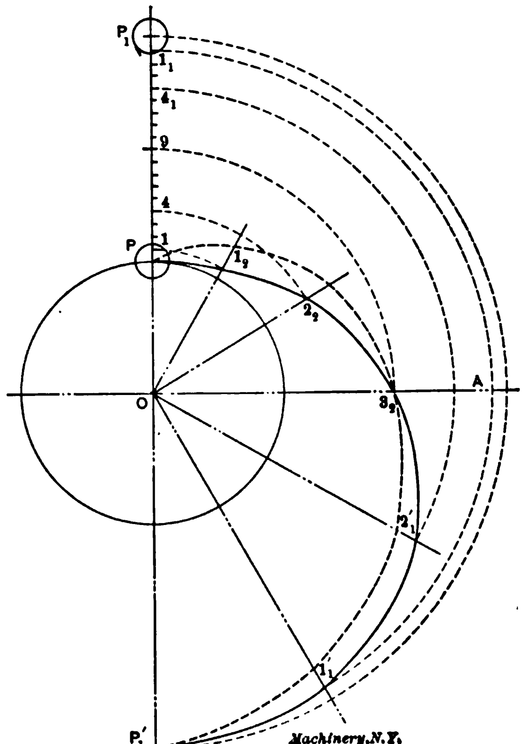


Fig. 25. Accelerated Motion or Gravity Curve applied to Plate Cam

uniformly accelerated and retarded. The sides of the groove are made parallel to $S T_1$, and drawn to suit the diameter of follower P.

Fig. 28 shows the distortion of the curve $S T$ when the follower moves in the arc of a circle, with center at some point Q, instead of in a straight line. Points on the new curve are found by setting off from the intersections b_1, d_1, etc., the ordinates $a b$ and $c d$. The curve $S a_1 c_1 T$ is then made the center line of a groove which will drive the hinged follower with the same variation in speed attained by the follower in Fig. 27.

Accelerated Motion Plate Cam

Fig. 25 shows how the parabolic curve is applied to a plate cam. The roller follower is supposed to oscillate between P and P_1 as the cam rotates about O. The curve $P3_2P_1'$ corresponds to $S\,T_1$ in Fig. 27, being the center line of the parabolic groove in the face of the plate.

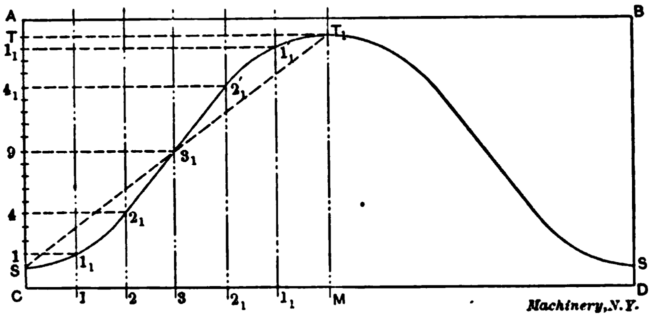


Fig. 26. Development of Uniformly Accelerated Motion Curve

Only one-half of the cam is shown in the figure. Suppose this cam is to rotate 180 degrees, while the follower moves from P to P_1. Draw the base circle with radius $O\,P$, the length of which will depend upon the size of the cam. Draw $O\,A$ perpendicular to $O\,P$, and divide the arc subtended by $P\,O\,A$ into any convenient number of parts, say three. Draw radii $O1_2$, $O2_2$, etc. Divide $P\,P_1$ into two equal parts at 9, and divide $P9$ into the square of three parts, or 9, as shown. With O as a

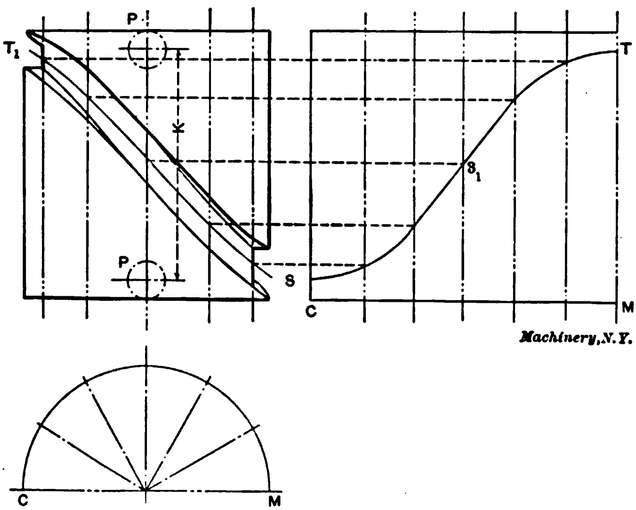


Fig. 27. Transferring Uniformly Accelerated Motion Curve to Cylinder

center, and radius $O1$, find the intersection 1_2. In the same way find the other intersections 2_2, 3_2, etc., and draw a smooth curve through these points. This curve has the same relation to the curve of uniform

motion shown dotted, that the parabolic curve has to the straight line in Fig. 26. If a similar curve be laid out on the other side of $P P_1'$, and made the center line of a groove, then the follower P will be pushed up and down mechanically by direct contact. If a curve parallel to $P3_2 P_1'$, and drawn at a distance equal to the radius of the follower away from

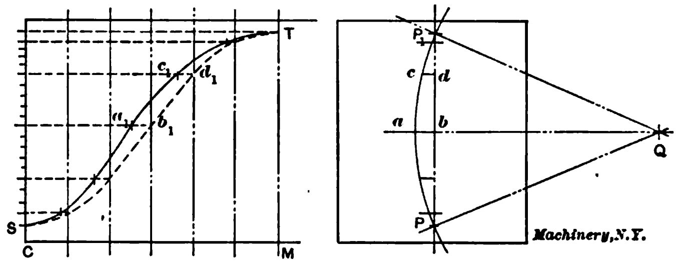


Fig. 28. Accelerated Motion Curve, when Follower moves in the Arc of a Circle

it, on the inside, be made the outline of the cam, then the follower will be pushed up mechanically to P_1, and allowed to fall by its own weight. It will remain in contact with the cam theoretically, because the principle of uniformly accelerated motion is the same as that of a falling body. In practice, however, the friction and the inertia of the connected

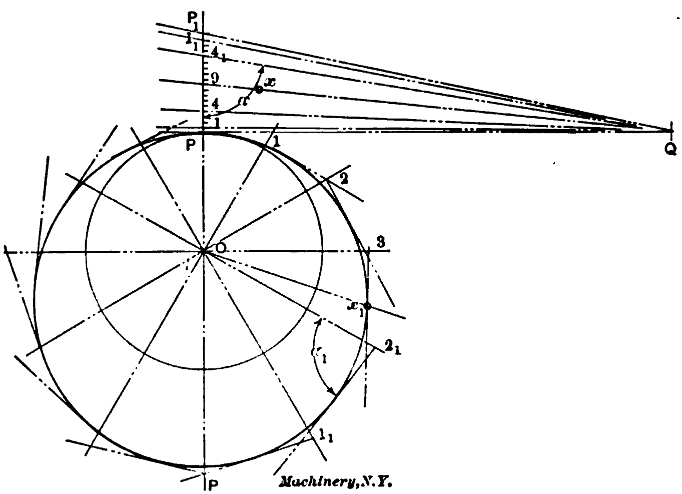


Fig. 29. Plate Cam for Bar Follower

parts would probably prevent the follower from remaining in contact with the cam on its return motion if the oscillations were rapid.

Fig. 30 shows the parabolic cam constructed for a follower which moves in any curved path. The construction is the same as in Fig. 25 except that points on the curve are located on radial lines Oa_1, Ob_1, etc., offset from the first radii by the distances $2a_1 = 4a$, $3b_1 = 9b$, and so on.

Plate Cam with Bar Follower

When a plate cam is to be laid out to drive a bar follower through a certain cycle of operations, the construction is more complicated. The base circle is divided as in the previous case into any convenient number of parts, and the square of the number of such parts laid out from P to 9 and from 9 to P_1, Fig. 29. If the bar is to oscillate about Q as a center, it will take the positions $Q1$, $Q4$, $Q9$, etc., as the radii $O1$, $O2$, $O3$, etc., come to the position OP. The intersections 1, 2, 3, and so on, are found just the same as in the previous cases. Now instead of drawing the curve for the cam outline through these points, straight lines which represent the edge of the follower must be drawn

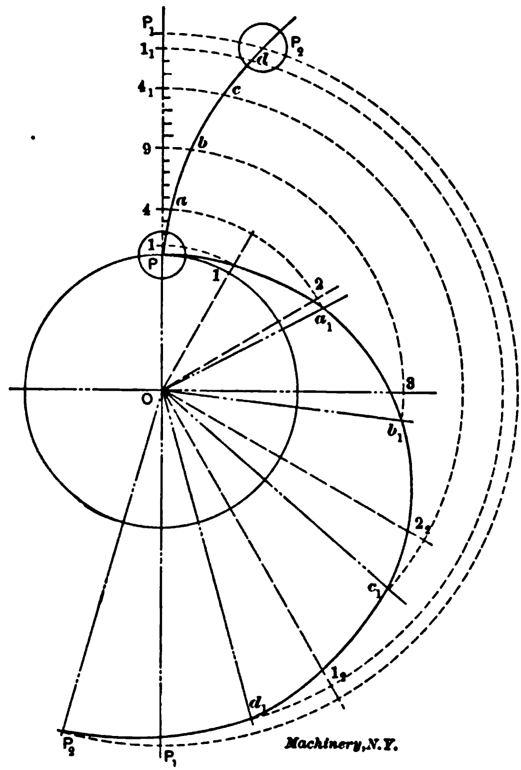


Fig. 30. Accelerated Motion Curve applied to Plate Cam, with Follower moving along a Curve

through the points making the same angle with a given radius as the follower makes with OP when the radius in question is in the position OP. For example, angle a equals angle a_1. Now the cam outline is a smooth curve drawn tangent to these straight lines. If the bar follower, instead of being centered at Q, moves up and down parallel to its first position, then all these angles are right angles. If the face of the bar is curved, then the cam outline must be drawn tangent to the

curves after they have been properly located with respect to their several radii.

In drawing cams like Fig. 29, the proper relation between the diameter of the base circle and the distance PP_1 must be assumed. If the base circle is too small, the cam outline will not be tangent to the edge of the follower in all positions, and the latter will not have uniformly accelerated and retarded motion. There is a rolling and sliding contact between the cam and its follower in the case of Fig. 29. The rolling action tends to carry the point of contact outward to the right of OP, during the upward motion, and to bring it back towards OP during the downward motion. The point of contact x does not necessarily occur when Ox_1, is perpendicular to Qx.

Effect of Changing Location of Cam Roller

When the line of motion of a follower passes through the center of rotation of the cam and the angle of the curve causes it to work hard,

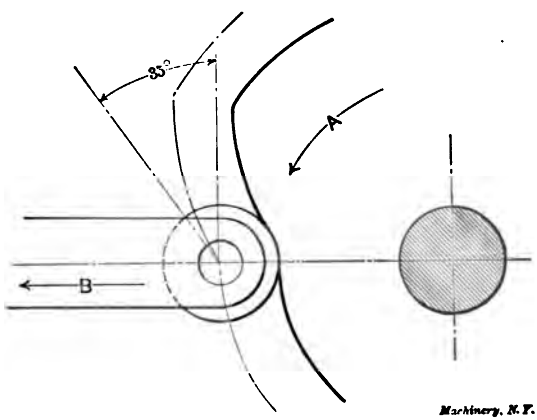


Fig. 31. Cam Roller on Center Line of Cam

the curve may be modified, and the same motion of follower obtained by placing the follower with its line of action parallel to its original position and not passing through the center of the cam. A condition may be assumed, as shown in Fig. 31.

Here we have a cam, rotating in the direction indicated by the arrow *A*, whose duty it is to move the follower ¾ inch in the direction indicated by the arrow *B* during a 30-degree angle of motion of the camshaft. The angle of the cam as presented to the follower at the beginning of the stroke would be 35 degrees, as determined by the tangent to the curve of the centers, as indicated on the drawing. After the follower had moved one-third of its distance, the angle presented would be 32 degrees, and when two-thirds of the travel had been made, the angle of the curve would be about 30 degrees. The angles given are for a curve which would give a uniform motion to the follower. Should the cam curve work hard at the required speed we would naturally make the cam of greater diameter, if possible, which would reduce the

angle of the cam, as shown by the difference in the angles presented in Fig. 31, as we go out from the center of rotation. The design of the machine, however, might make this change impossible. If it was simply necessary to get the follower from the position shown to a point ¾ inch distant in a 30-degree movement of the cam-shaft, without regard to its motion, a harmonic or gravity curve might be used which would cause the cam to work easier. However, this would be impossible should our design require a uniform, or some other equally hard motion. A third way in which the angle of the curve might be decreased would be to make the angle of motion of the cam-shaft greater. This, too, might be made impossible by the limitations of our design.

Another way, and one not commonly used, consists in changing the location of the cam roller. In Fig. 32 all conditions are the same as

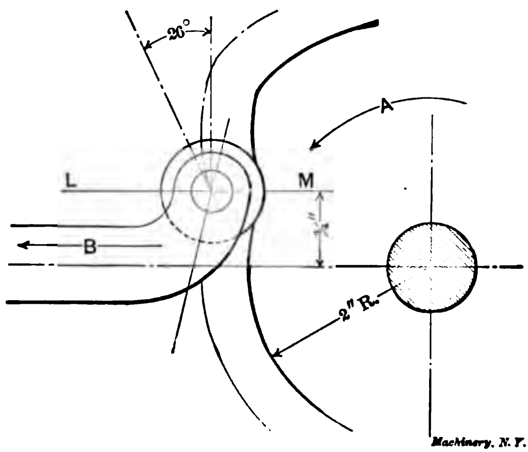


Fig. 32. Cam Roller placed above Center Line of Cam

in Fig. 31, except the roller has been placed ¾ inch above the line passing through the center of the cam. The center of the roller will now pass along the line *L M*, or parallel to the line of motion in Fig. 31. The angle of the curve presented to the roller in this case is 26 degrees, much less than the angle presented in Fig. 31, and the angle decreases as the roller moves away from the center of rotation. The advantage that may be gained by moving the cam roller may be readily seen by comparing the results given above. There is, of course, a limit to the distance the roller may be changed, for if placed too far away from the center line, the thrust in the direction at right angles to the direction of motion of the follower would be so great as to offset the advantage gained.

Even without the aid of an illustration it may be seen that to place the cam roller on the other side of the center would cause the angle of the cam curve to increase, thus making conditions worse. The offset of the roller should be in the direction opposed to the direction of motion of the cam.

CHAPTER III

NOTES ON CAM DESIGN AND CAM CUTTING*

It is strange that the processes and methods of cam cutting have not been improved more rapidly than they have. Twenty-five years ago, cams and gears were on about an equal footing; that is to say, most of both were cast to as nearly the proper shape as possible, after which the working surfaces or teeth were smoothed up with a file, and then the holes and hubs were finished in the usual manner. Some cams of both plate and barrel forms were cut, with suitable attachments, in the same machine the gears were cut in. This was an old hand indexing machine, with an automatic feed composed of a weight hung on the pilot wheel. Since that time gear cutting machinery has been wonderfully developed. All sorts of styles and arrangements are on the market, meeting every demand, from that for a general purpose machine to highly specialized forms. When it comes to cam cutting machinery, however, while machinery builders have special tools for their own work, so far as the writer is aware, there is no tool regularly on the market for cutting cams. The cam has thus fallen behind the gear in the process of development. Machine designers and machine users are liable to be a little suspicious of cams, anyway. Considerable trouble is often taken to avoid the necessity for using them. This is due, however, as much to faulty design and faulty construction as to any inherent objections to this form of mechanical movement. It is here proposed to call attention to some of the points to be considered in designing and producing satisfactory cams, with the thought of thereby doing something to justify a more extensive use of them.

Faults in the Design of Cams

We have all seen cams that were the cause of a good deal of profanity, in which the trouble could be traced to the designer or machinist, who laid out the curves on what might be termed "schedule time"; that is to say, he simply made sure of his starting and stopping points, neglecting all intermediate points so long as the movement got there and got back on time. This, he thought, would be all that was necessary, not taking into account the shock and jar caused by the sudden starting and stopping of heavy slides, levers, etc., at even moderate speeds. The temptation to do this is always strong, especially in the case of barrel cams, where it is so much easier to use the milling machine (gearing it up for a spiral to meet the schedule requirements), than it would be to lay out and form a curve with a gradual starting of the motion and a gradual stopping. There is nothing worse for the life of a machine than to have it operated by cams cut by this "sched-

* MACHINERY, August, 1907.

ule" method. Another point to consider is that of taking advantage of all the time there is for any given movement. The period or periods of rest should be cut down to the last degree, so as to have the angularity of the rise as small as possible. Careful work at the drawing-board will make a big difference with the satisfactory action of cams in these two respects. Still another bad practice, which has perhaps tended to throw the use of cams into disfavor, is that of making them in two or more parts, with the idea of having the working surfaces adjustable. After they have been wedged out, or shimmed up, or ground off a few times, a more proper name for them would be "bumpers" rather than "cams." Except in rare cases, there is no more use or excuse for adjustable cams than for adjustable gears, as there are other and better means of making adjustments when these are necessary. Cams are not very expensive as compared with gears, and they can be duplicated with greater accuracy than most machine parts. Especially is this the case if roughing and finishing mills are used in forming them, as the finishing mill will retain its cutting edge and size for a great number of cams, if it runs true with the spindle in the first place.

Cam Rolls and Roll Studs

A few words might be said with relation to the design and construction of cam rolls and the studs for them, since the successful working of a cam depends to a considerable degree on this matter. The design of the roll and its stud should be such that the work it has to do, the speed at which it runs, and the bearing area on the stud, should be the factors determining its size, rather than the simple fact that there is a milling cutter in the tool-room of a certain diameter. It is equally important that the roll and stud should be ground all over after hardening. The end of the roll should also be cut back for 1/64th of an inch or so on the sides for some distance from the outside diameter, so as to avoid undue friction against the collar of the stud, or the part it is mounted in. On account of the warping that takes place in hardening, rolls that are not ground inside and out have a habit of stopping frequently under load, until in time flat spots are worn on the face; then the working surface of the cam will begin to wear or rough up. Roll studs that are the slightest degree out of parallel to the working surface of the cam will also cause some trouble, but no amount of grinding will help this case. The same trouble occurs on barrel cams if the milling cutter is set above or below the center of the cam when cutting it. The roll will then bear at one end only at the most important time, when the throw takes place. A conical roll is the proper thing for this style of cam. There is a lot of end pressure to a roll of this type, however, which must be taken care of by thrust collars on the stud; or, better still, a ball race may be scored in the collar and the large end of the roll, so as to provide for a ball thrust bearing. This end pressure will reduce the side pressure on the stud to quite an extent, nevertheless, so the latter may be made slightly shorter or smaller in diameter than when a parallel roll is used.

Cutting Cams of Uniform Lead in the Miller

When it comes to the cutting of cams, the shop man naturally turns to the milling machine. Many manufacturers of milling machines make attachments which may be used for cutting cams with formers. None, however, is provided with anything except hand feed. Another, and the greatest, objection to them is that if there is much work to be done, one of the most expensive machines in the shop is tied up, and there are few shops that have a surplus of this brand of machine tools. For an occasional or an experimental job, however, there is nothing better than the milling machine. As has been before remarked, curves with easy starting and stopping movements cannot be cut

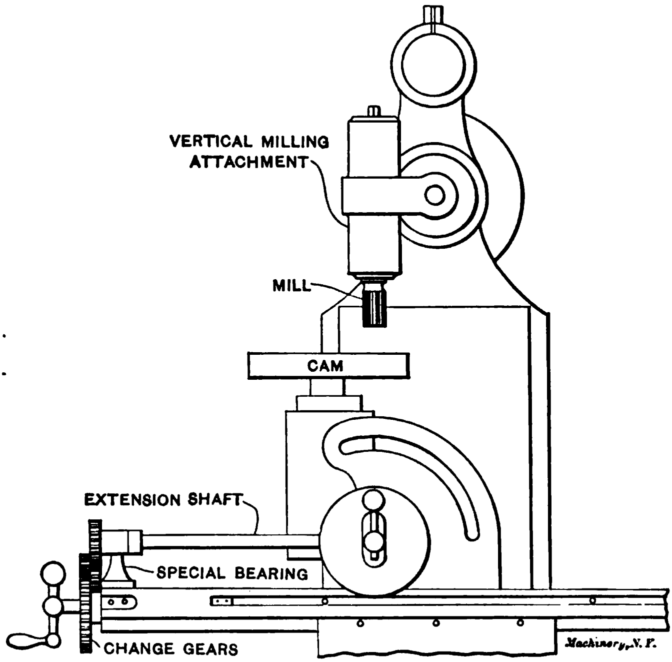


Fig. 33. Cutting a Face Cam of Uniform Rate of Throw

without formers on it, or on any other machine for that matter; but cams which require a constant rise, such as the feed cams of some machines, may be cut on it without the use of formers. With barrel cams the method is obvious, it only being necessary to gear the spiral head with the lead-screw to get the required lead, and then cut a groove of this pitch in the body with an end mill of the same diameter as the roll.

For cutting plate cams for the same kind of motion, the arrangement shown in Fig. 33 may be used, if the machine happens to have a vertical spindle milling attachment and a spiral head. All that it is necessary to provide in addition is the extension shaft shown, and the special bearing or bracket for supporting it. These parts are used

to bring the spiral head to the center of the table. The shaft is bored out at one end to fit the stud of the spiral head (called the worm gear stud in the tables); the other is turned and keyed to fit the change gears. The cams may be held in the regular chuck, or on a face-plate fitted to the head. Small ones may be held on an arbor fitted to the spindle, with large collars to hold them firmly, clamped with a nut and washer, or by an expansion bushing in the case of large holes. If they have keyways in them, and more than one or two are to be made, it will be well to fit a key in the arbor to help locate them. It is necessary to set the mill central with the spiral head to obtain correct results, as the spiral will vary if this is not done. Advantage may sometimes be taken of this when, with the regular change gears, there is no spiral of the exact pitch required, in which case the desired rise

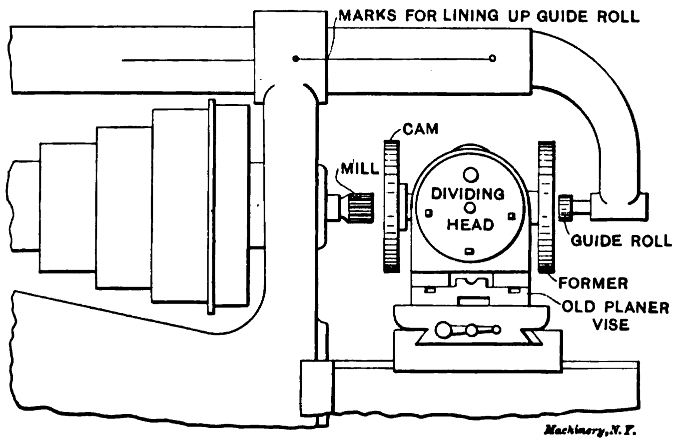


Fig. 34. Inexpensive Fixture for Milling Plate Cams to Match a Former

can be obtained by setting the head off center. This, however, will not give a uniform spiral, as the pitch will keep increasing as it leaves the center of the cam. As cam drawings are generally laid out or divided in degrees, it will be found convenient to divide the cam blank by the same method, while held in the spiral head. To do this, we may revolve the index crank through two holes in the 18-hole circle or three holes in the 27-hole circle, as many times as are necessary, each of these divisions giving exactly one degree.

Milling Machine Attachments for Cutting Cams with a Former

Examples of attachments rigged up to suit special requirements are shown in the cuts Figs. 34 and 35. To a shop with a rather limited equipment, an order came in for a lot of eight machines, which required seven cams each, most of which were of the plate type. As this class of work was new to the shop, there were no facilities for this part of the job; as usual, it was decided to do the work on the milling machine.

An old planer vise was scraped up and refitted so as to have the

movable jaw a nice sliding fit—the screw having been removed, of course. To this jaw was fitted and bolted the spiral head of the miller, in such a way that its spindle could be placed either at right angles, or parallel to the cutter, as the case required for barrel or plate cams. An arbor was made, long enough to pass through the head, carrying the former on the back end and the cam blank on the front end. A nut threaded onto the back end held the former against the end of the spindle, so there was no danger of the arbors rattling loose, no matter how badly the work and tool chattered.

For plate cams, as shown in Fig. 34, the former was made the opposite hand to that of the cam required. The overhanging arm had a center line marked on it as shown, which was matched with one on the frame so as to locate the arbor support central with the spindle. In the place of the arbor-supporting center there was fitted a stud

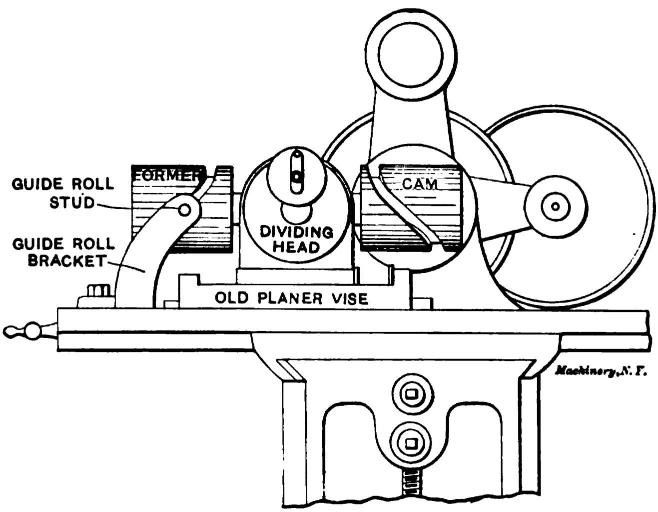


Fig. 35. Cutting a Cylindrical Cam with the Attachment shown in Fig. 34

with a roller of the same diameter as the cutter. The arm was held securely by the regular milling machine braces, which are not shown in the cut. The method of operation is obvious. The spiral head with its attached work and former was revolved, slowly, by hand. The action of the roller, held by the overhanging arm in the groove of the former, caused the head and work to slide back and forth on the ways of the planer vise, giving the proper movement between the work and the cutter to produce the desired contour of cam. The table was locked on the saddle.

For barrel cams, the attachment was rearranged as shown in Fig. 35. The former roller was held firmly in a bracket bolted to the table of the machine. As the roller is on the opposite side of the milling cutter, the former and work are set 180 degrees apart on the work arbor, otherwise they are alike. The head is relocated on the movable vise jaw to bring the axis of its spindle at right angles to the axis of

the cutter, as shown. The reader will easily make out the other details from the engraving.

Both arrangements cut good cams, considering that the first cost of the whole outfit was very little. As the formers were made accurately to drawing, the cams gave good satisfaction at fairly high speeds, but the device had the disadvantage of tying up a machine which had plenty of work waiting for it; besides, it was a tedious job to feed the index crank by hand all day long, especially when working on steel cams. For these reasons, when a duplicate order came in, a few weeks later, it was considered best to try the plan of cutting the plate cams on an old lathe, thus providing the advantage of an automatic feed, and relieving the miller of some of its work as well.

A Face Cam Cutting Attachment for the Lathe

A lathe cam cutting attachment is shown in Fig. 36. While not new in principle, it differs somewhat from the other makeshifts described.

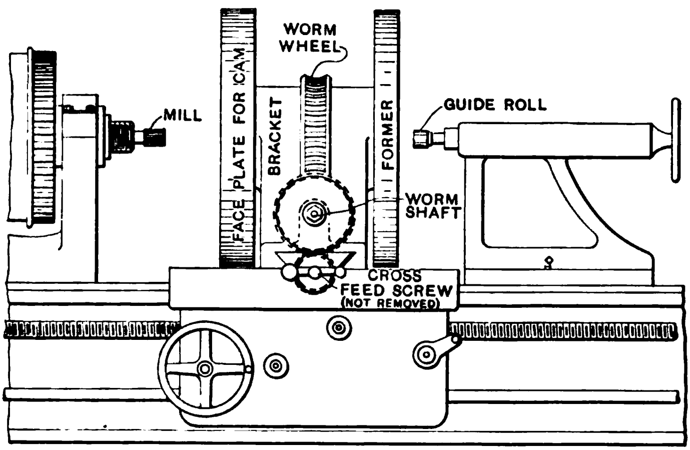


Fig. 36. Attachment with Power Feed for Cutting Face Cams

For this arrangement, the tool slide was removed from the machine and replaced with the bracket casting shown. This was fitted and gibbed to the tool-rest slide, and had its spindle bored and sides faced with a boring bar on the lathe centers. To the bracket was then fitted the cam face-plate and spindle, cast in one piece and finished all over, with the back or small end threaded to fit the former. Keyed to this spindle was a worm-gear of cast iron. In this case the worm-gear had 82 teeth. Meshing with this gear was a worm having 9/16 inch hole, and with a key having a sliding fit in the worm shaft. Bearings were provided for the worm shaft at front and back. The front support for the worm shaft was cast onto the bracket, and finished with it to fit the tool-rest slide, after which it was sawed off and fastened at the front of the carriage by the gib screw, as shown. This is the same practice as is commonly followed in making the clamp for the thread-

ing stop on the cross slide. To the outer end of the worm shaft was keyed a gear, meshing with another fitted and keyed to the front end of the cross feed screw next to the handle. The quill was cut off to make room for it. The cross feed nut was removed entirely, of course.

It will be seen that this arrangement, while having the general features of that shown in Fig. 34, provided the advantage of making use of a less costly and less over-worked machine, and allowed the use of a power feed as well, since the gearing provided for connection with the power cross feed in the apron. This gave a feed fine enough for small cams, but on large ones it was necessary to run the feed belt from the feed shaft cone to the hub of the large intermediate gear of the screw-cutting train, this being in mesh with the spindle gear. The lead-screw was removed so as not to interfere with the belt. With regular changes this gave a wide range of feeds.

The cams and formers were held to their respective face-plates by bolts. All the formers were of the positive follower type having a groove for the guiding of the roller. No weight or other means is then required for the followers to hold them to their work.

CHAPTER IV

CUTTING MASTER CAMS*

Common Method of Making Master Cams

Assuming that the master cam has been properly machined and roughed down, we will consider briefly the generally used method of finishing it. This method comprises mounting the master cam in the dividing head of a universal milling machine, and gearing the head with the feed-screw of the table so that the table will advance in proper ratio with the turning of the work in the dividing head. In Fig. 37 a master cam is mounted as above described, and held against a cutter in the vertical spindle milling attachment on a milling machine. This cutter is of the same diameter as the roll which will be used with the cam. The following description refers specifically to the cutting of the master cam for the cam shown in Chapter I, Fig. 17.

The process is as follows: Feed the work against the cutter until

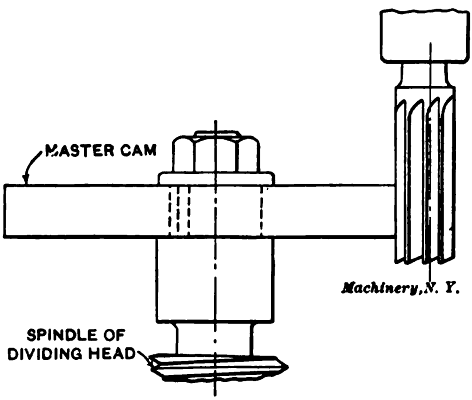


Fig. 37. Common Method of Milling Master Cams

the cutter is 3 11/16 inches from the center of the master cam. Now, with the key-slot of the master cam which is the "zero" of the cam, directly in line with the cutter, turn the work 150 degrees. This finishes a part of the outer dwell of the cam. The next operation is to feed the work against the cutter ½ inch while the dividing head turns 45 degrees. Since 45 degrees is ⅛ of 360 degrees, or one turn, we want gears which will turn the work ⅛ of a revolution while the table advances ½ inch. This is equal to one turn of the work while the table advances 4 inches. The gears on a feed-screw with four threads per inch, and 40-tooth worm-gear in the dividing head are:

Gear on worm 36,	Second gear on stud 28,
Gear on worm 36,	Gear on screw 70.

* MACHINERY, October, 1908.

Having connected these gears with care, feed the work against the cutter 0.500 inch. The gears will at the same time turn the work 45 degrees. This will give the advance of the cam. Now, with the table clamped where it is, turn the work 35 degrees further. This will give the inner dwell of the cam. Now change the gears so that the work will turn 90 degrees while the table is backed away ½ inch. This may be done by removing the first gear on the stud with 36 teeth and replacing it with a 72-tooth gear. Having done this with care to avoid disturbing the work during the change, back the work away from the cutter 0.500 inch. The gears will have turned the work 90 degrees more, the intermediate having been properly adjusted. This will give

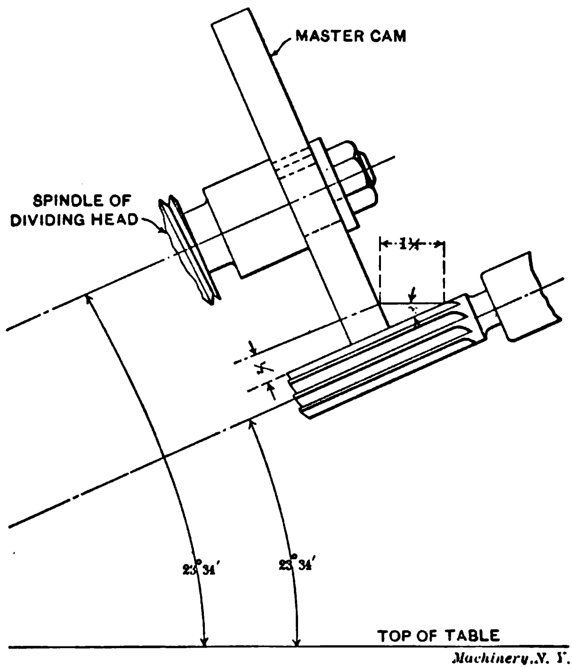


Fig. 38. Improved Method of Milling Master Cams

the retreat of the cam. Now, with the table clamped where it is, turn the work until the cutter reaches the part already finished.

The method which has just been described, is very convenient when the change gears will give the combinations that are necessary, but it often happens that the desired combination cannot be made with even an approach to accuracy. This difficulty may be overcome, however, by a method which is not in general use, but by which any desired result may be obtained.

Improved Method for Producing Master Cams

For convenience we will suppose that the master cam could not be cut with the gears named or with any others, in the vertical position.

We will proceed as follows: Mount the roughed-out master cam as before in the dividing head, and place a 1-inch end mill in the vertical milling attachment, but, instead of setting them in a vertical position, incline each at an angle of 23 degrees 34 minutes, as shown in Fig. 38. The reason for this will appear later.

By inspection we see that if the work be fed against the cutter, Fig. 38, the cutter will enter the work and approach the mandrel. We also see that if the angle of inclination be increased or reduced, the

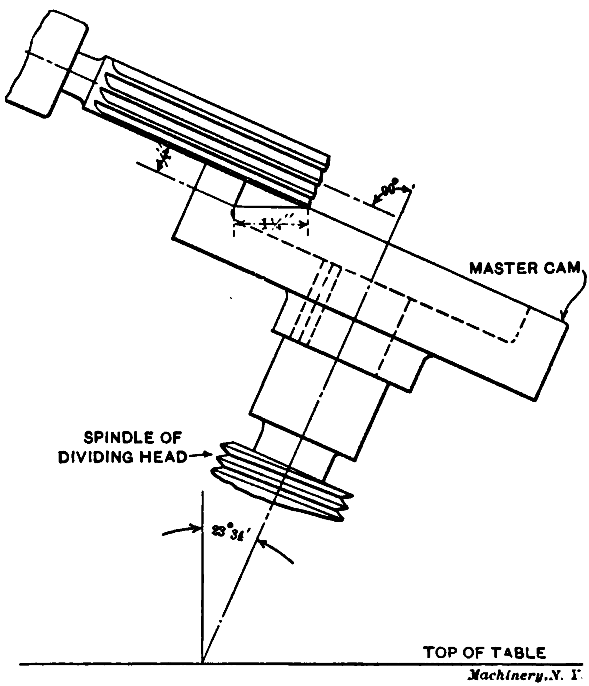


Fig. 39. Milling a Master Cam for a Drum Cam

rate with which the cutter approaches the mandrel will vary likewise. A convenient combination of gears to use in this case is one which will turn the work 360 degrees while the table advances 10 inches. This result may be obtained by using four 36-tooth gears to turn the work.

Having milled the master cam for the first 150 degrees to a radius of 3 11/16 inch as mentioned, we must find the correct distance to feed the table forward in order to make the cutter approach the mandrel ½ inch while the work turns 45 degrees. The computation is done as follows: Forty-five degrees is ⅛ of 360 degrees. Since the table is geared to advance 10 inches while the work turns 360 degrees, the table will advance ⅛ of 10 inches while the work turns 45 degrees. Thus the advance is 1¼ inch to the 45-degree turn of the work. By inspection we see that in Fig. 38 the cutter and the work-face form two sides in a right-angled triangle with a hypothenuse of ¼ inch

and one side of ½ inch. By solving, we find the angle a to be 23 degrees 34 minutes, as before mentioned. Having now properly connected the gears to mill the advance on the cam, feed the table ahead 1.250 inch. As just stated, this will make the cutter approach the mandrel ½ inch while the gears will have turned the work 45 degrees. Now with the table clamped where it is, turn the work 35 degrees more. We are then ready to begin the retreat of the cam. We must arrange gears which will turn the work 90 degrees while the table is backed 1¼ inch. By removing the 36-tooth gear from the screw and replacing it with a 72-tooth gear, we get this result. Carefully make the change so as not to disturb the work, and then back the table 1.250 inch. The gears will have turned the work 90 degrees further. Now, with the table clamped where it is, turn the work until the master cam is completed.

This system for making cams may be used only where uniform movements are required. While we have used it to entirely finish a master plate cam, any part of any cam requiring uniform motion may be

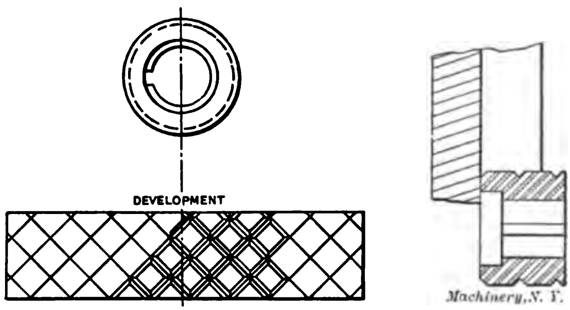


Fig. 40. Special Finishing Cutter for Cam Grooves

milled in this way with a degree of accuracy not readily obtained in any other way. In fact, the work should be as true as the machine on which it is done. The same system may be used to make a master cam for a drum cam, as shown in Fig. 39. Note, however, that the work is set 23 degrees 34 minutes from the vertical position, while the cutter inclines at right angles to, instead of parallel with, the axis of the mandrel. The same combinations of gears would be used if the drum cam action were similar to the one which we have discussed. The exceedingly low cost of making master cams by this method makes it profitable to provide a master cam for cutting the groove in a single cam.

Special Cutter for Finishing Grooved Cams

A source of constant annoyance in milling grooves in cast iron cams lies in the fact that finishing cutters quickly wear and become under size. They must then be laid aside or used for taking the roughing cuts, while a new cutter of full size is used for finishing. We will not discuss the practice of putting a piece of paper in the collet to make the small cutter run out of true. Another source of trouble, even

with cutters with spiral flutes, is the tendency of the cutter to chatter, unless it is perfectly ground and all other conditions are exactly right. Still a third trouble is in the tendency of the cutter to cut more on one side than on the other and to dig out stock in spots in the groove.

In Fig. 40 is shown an extremely simple tool, the usefulness of which cannot be overestimated for finishing grooves in cast iron cams. It is a piece of tool steel, suitably machined to mount on an arbor. It is turned on the outside, with enough stock left on for grinding, after which the spiral grooves shown in the developed surface are milled with an angular cutter. The piece is then hardened and ground to size. The cam groove which we are to finish is roughed out from 0.002 inch to 0.012 inch below size; the roughing cutter is removed from the spindle of the cam cutting machine, and this special tool is mounted in its place. The cam is then fed against the tool until the tool reaches the bottom, when the cam is turned one complete revolution. The tool will leave a true groove exactly the right size, and without chatter marks or hollows.

By reason of the form of the cutting or scraping edges, it will outlast many ordinary cutters. Used in connection with it, a single roughing cutter may be repeatedly sharpened before it becomes too small for good results.

CHAPTER V

SUGGESTIONS IN CAM MAKING

In the present chapter are collected a number of suggestions for the laying out and making of cams, together with a discussion on the shape of cam rollers for cylinder cams. These suggestions have been contributed from time to time to the columns of MACHINERY. The names of the persons who originally contributed the matter here selected, have been given in notes at the foot of the pages, together with the month and year when these articles appeared.

Making Master Cams

The method of originating cylindrical master cams, which is described in the following paragraphs, has been used successfully in a shop where considerable of this work is done. A development of the

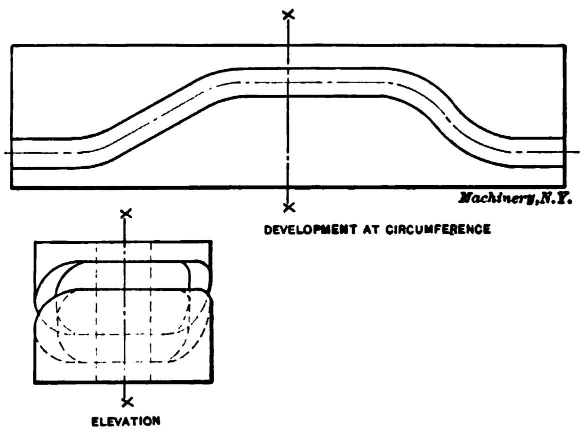


Fig. 41. Master Cam and its Development

cam at the surface of the cylinder is provided by the draftsman. If the cam is smaller than 2½ or 3 inches diameter, or has unusually steep pitches in its make-up, the development should be laid out for a diameter two or three times larger than that of the desired cam.

Suppose it is desired to make a master for the cam shown in Fig. 41. The first step is to make a template to match the development shown in the drawing. This template may be made of mild steel, of a thickness depending upon the diameter to which it is to be bent, as described later. It may be fitted to the drawing with cold chisel and file, or, if considerable accuracy is desired in the throw, the template may be held in the milling machine vise, and the straight surfaces finished to the graduations. This template, shown in Fig. 42, is made for one side of the cam groove only.

The next step is to turn up a piece of steel or cast iron, as shown at *B*, Fig. 43, to such a diameter that when the template *A* is wrapped around it, as shown, the ends will just barely meet. This diameter is about the thickness of the plate less than the diameter to which the development was laid out, but it should be left a little larger and then fitted. The plate is now clasped around the body, with the back edge pushed close up against the shoulder to insure proper alignment

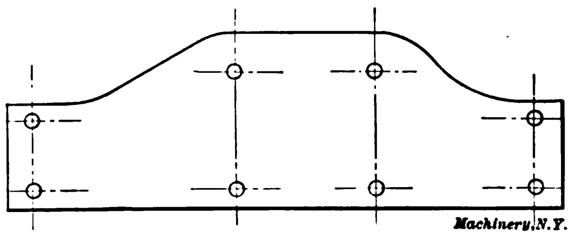


Fig. 42. Template for Making Master Cam in Fig. 41

of the working surface of the cam. If any difficulty is experienced in this wrapping process, a circular strap may be bent up with projecting ends as shown in dotted lines at *C*; with the aid of a clamp *D* the template may be stretched around smoothly. The template and the body may now be drilled and tapped for screws, as shown, and for dowels as well, if found necessary.

Scribe the contour of the cam onto the body *B*, remove the template, place the body on an arbor between the index centers of the milling machine, and take away the stock for about 1/8 inch deep, or so, 1/16

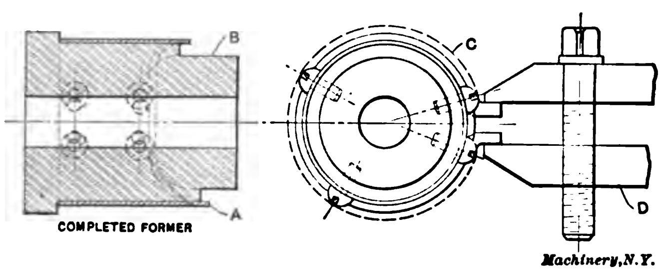


Fig. 43. Template Secured on Mandrel for Making Former

inch back of the scribed line. This, as shown in Fig. 43, is for the purpose of providing a clearance underneath the working edge of the template. The template may now be placed in position on the body once more, and fastened there. The arrangement is now ready for use as a former for making a master cam.

Fig. 44 shows a milling machine arranged for cam cutting. *E* is a casting made to grip the finished face of the column, and carrying an adjustable block *F*. Cam roll *G* is pivoted on a post which is adjustable in and out in block *F*. Our former *H*, and master cam blank *I*, are mounted, as shown, on an arbor between the index centers. By working the index worm crank, and the longitudinal feed together,

roll *G* may be made to follow the outline of former *H* in such a manner that the end mill will cut the desired groove in cam blank *I*. A slightly smaller mill may be used for a roughing cut, but it goes without saying that the roll and the finishing mill must be of about the same size if a true copy of the template is desired. It will be found

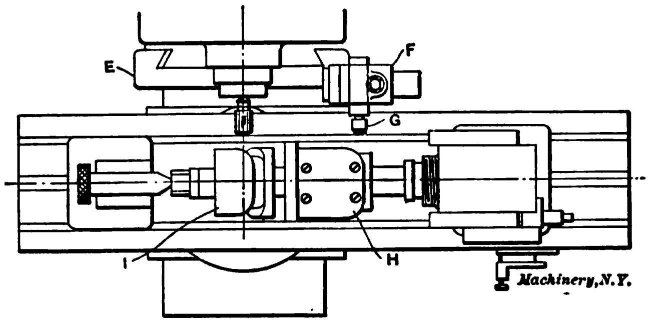


Fig. 44. Arrangement of Milling Machine when Using the Former

easier to follow the outline with the roll if the steeper curves are traced down rather than up.

A fairly good cam cutting machine for making copies of the master cam *I* may be improvised by using the attachment *E*, *F*, *G* in a rack feed machine. It might also be feasible to connect the index worm

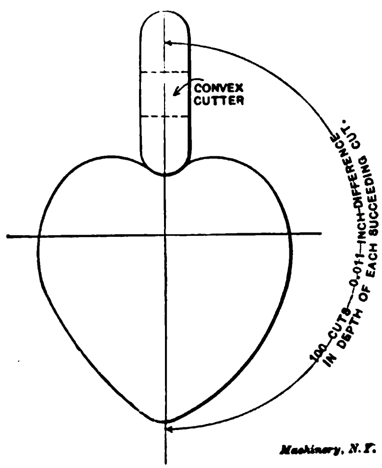


Fig. 45. Method of Cutting Cams

with the telescopic feed shaft so as to give a power feed to the contrivance. To insure accurate cams, the arrangement for holding the tool must be made stiff enough to move the table without much spring, or the table must be weighted, so as to bring the pressure of the roll constantly against one face of the master cam.*

* R. E. Flanders, July, 1904.

Simple Method for Cutting Cams Accurately

Cams are generally laid out with dividers, machined and filed to the line. But for a cam that must advance a certain number of thousandths per revolution of spindle this divider method is not accurate. Cams are easily and accurately made in the following manner. For illustration, let us make the heart cam in Fig. 45. The throw of this cam is 1.1 inch. Now, by setting the index on the miller to cut 200 teeth and also dividing 1.1 inch by 100 we find that we have 0.011 inch to recede from the cam center for each cut across the cam. Placing the cam securely on an arbor, and the latter between the

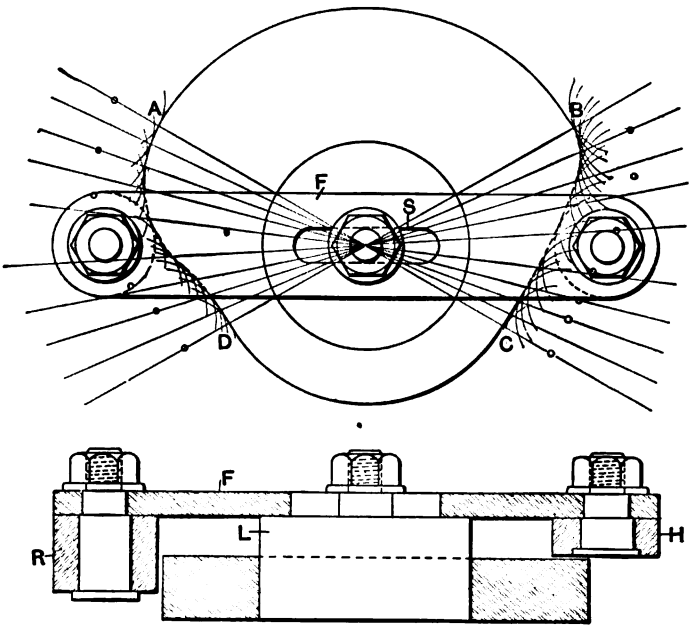


Fig. 46. Device for Correctly Laying Out Cams for Cam-Actuated Press

centers of the milling machine, and using a convex cutter, set the proper distance from the center of the arbor, we make the first cut across the cam. Then, by lowering the milling machine knee 0.011 inch and turning the index pin the proper number of holes on the index plate, we take the next cut and so on. Each cut should be marked on paper so that there will be no mistake as to number of cuts taken; when 100 cuts have been made the knee must be raised in order to complete the opposite side of the cam.*

Device for Laying Out the Cams of a Cam-Actuated Press

The cams which actuate the cutting or drawing slide of a double acting cam-press are different from other cams, inasmuch as each one

* F. E. Shailor, March, 1907.

actuates two rollers which are a certain fixed distance apart from each other. In order to avoid back-lash or springing of the connecting-rods, a fault which is to be found in most cam presses, it is evident that the rollers must both touch the face of the cam at all times. In Fig. 47 is shown the ordinary method of laying out such cams; this cut also shows the fact that this ordinary method does not accomplish the end desired. We see that in this cam both curves which give to the slide its up and down motion are constructed with the same radii, which clearly must give a curve that is faulty at certain points. The one main feature that our cam must possess can be expressed as follows: Two rollers of equal diameters, which are a certain fixed dis-

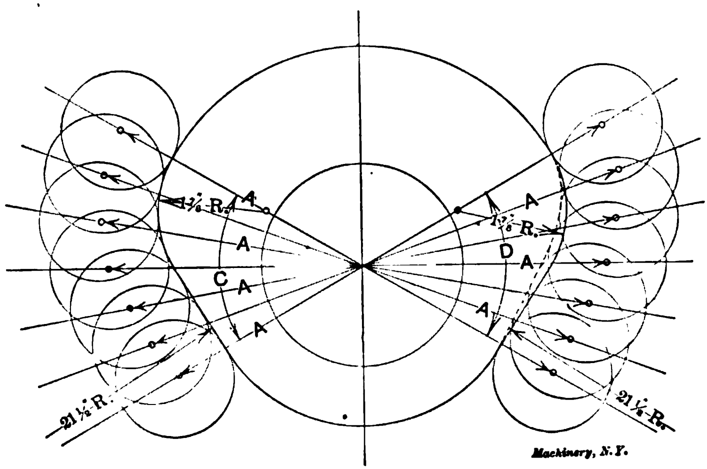


Fig. 47. Ordinary Method of Laying Out Cams

tance (*A* in Fig. 47) apart, on a line passing through center of cam, must always tangent the cam while the cam makes its revolution. Turning to Fig. 47, we see that the curve which spans angle *C* and the dotted curve which spans angle *D* accomplish this object. A little reflection will convince one that this curve cannot be constructed absolutely correct by giving the radii for both the up stroke and down stroke curve, owing to the fact that the shape of one is entirely dependent on the shape of the other.

We can, however, give the radii for one curve, and construct the other curve from it by the aid of the following device. It is assumed that in most cases it will be economical to cut a master cam, and use this for cutting the others. However, where only a few cams are to be cut, it will be well to construct one with the aid of our device, and use this one as a template for the others. Fig. 46 shows the device mentioned. First, cut the two arcs, *A B* and *D C*, which of course are perfect circular arcs of given radii, and also cut the curve *A D* from given radii. Then place center plug *L* into center hole of cam and fasten bar *F* onto *L*. Bar *F* has two rollers, *R* and *H*, fastened in such a way that their center distance is equal to the center distance of

the cam rollers in the cam press in which the cams are to be used. The rollers *R* and *H* have the same diameter as the cam rollers in the press. We now keep the roller *R* against the cam along the curve *A D* and follow this curve along its entire length. Center plug *L* will always keep the line connecting *R* and *H* in the center of the cam, and slot *S* enables us to follow the curvature of *A D*. By scratching the outline of roller *H* on the cam blank at very short distances apart, we will have a full outline on the cam blank, which must indicate the absolute curvature of *B C*. This curvature must possess all the qualifications previously set forth as absolutely indispensable for a correct cam-press cam. A cam or set of cams laid out in this manner will silence one of the principal objections usually raised against a cam-actuated press, *viz.*, back-lash or springing of the cam roller connecting-rods.*

Shape of Rolls for Cylinder Cams

The grooves and rolls for cylindrical cams are made in various ways, more or less suitable for the work to be done. Fig. 48 shows a

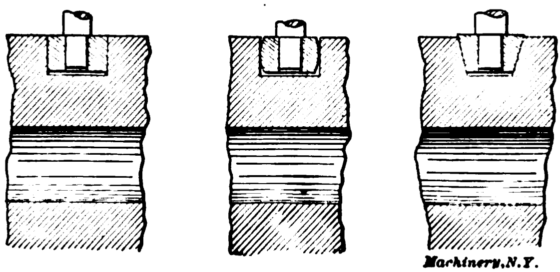


Fig. 48 Fig. 49 Fig. 50

Different Forms of Cam Rolls

straight roll and groove, Fig. 49, a roll with a rounded surface in a straight sided groove, and Fig. 50 a beveled roll and groove. In Fig. 48 the action of the roll is faulty, because of the varying surface speed of the cam at the top and bottom of the groove, due to its varying radial distance from the center line. This causes excessive wear and friction, especially in a quick running cam with steep pitches. For such cases, if the duty is light, the arrangement shown in Fig. 49 is better, as the roll has but a very small bearing surface, and is thus unaffected by a varying radial distance. For heavy work, however, the small bearing area is quickly worn down, and the roll presses a groove into the side of the cam as well, destroying the accuracy of the movement, and allowing a great amount of back-lash.

In Fig. 50 the conical shape is given to the roll with the idea of giving it a true rolling action in the groove. In most cases where this shape is used, however, the lines of the sides of the roll appear to converge to the center line of the cam, as shown in the figure. If the groove were a plain circumferential one, it would give a perfect action, like that of the pitch cones of two bevel gears rolling on each

* E. E Eisenwinter, July, 1907.

other. If the motion were in a line with the axis of the cam, without any circular movement, conditions would be perfect in Fig. 48. It is evident that in intermediate conditions, the groove must be given a shape intermediate between the two. In many cams of this variety the heavy duty comes on a section of the cam which is of nearly even pitch and of considerable length. In such a case it is best to proportion the shape of the roll to work correctly during the important part of the cycle, letting it go as it will at other times.

In Fig. 51, *b* is the circumferential distance on the surface of the cam, which includes the movement we desire to fit the roll to. The

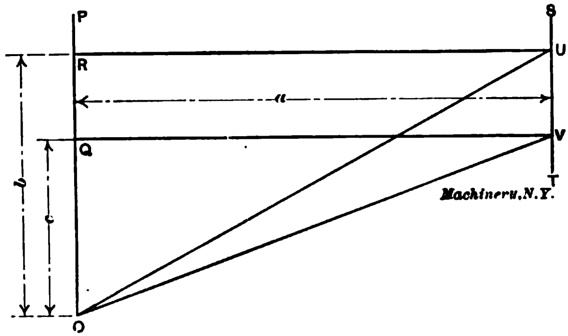


Fig. 51. Diagram Showing Method of Finding Shape of Cam Rolls

throw of the cam for this circumferential movement is *a*. Line *OU* will then be a development of the movement of the cam roll during the given part of the cycle, and *c* is the movement corresponding to *b*, but on a circle whose diameter is that of the cam at the bottom of the groove. With the same throw *a* as before, the line *OV* will be a development of the cam at the bottom of the groove. *OU* then is the length of the helix traveled by the top of the roll, while *OV* is the amount of travel at the bottom of the groove. If then the top width and the bottom width of the groove be made proportional to *OU* and *OV*, the shape will be suitable to give the result we are seeking.*

* R. E. Flanders, December, 1904.

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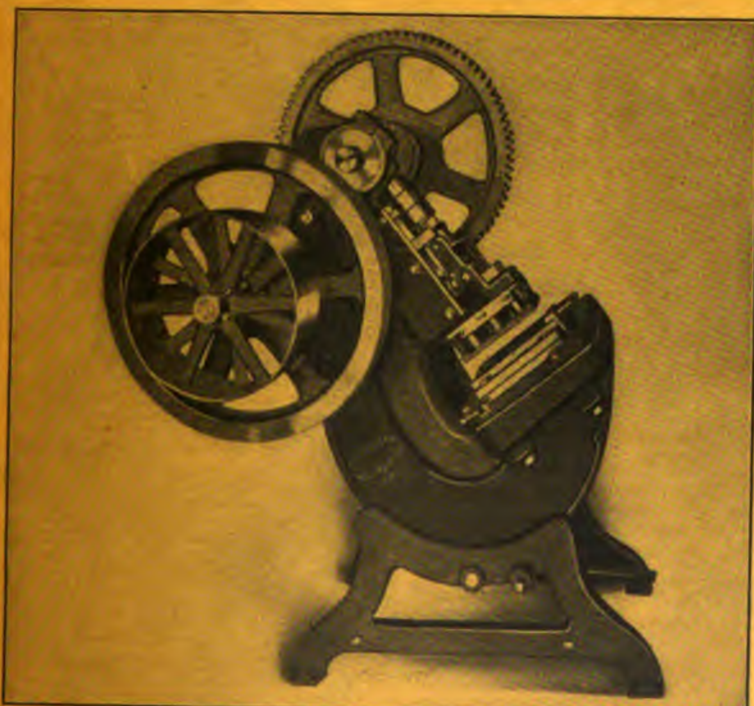

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NUMBER 13

BLANKING DIES

THIRD EDITION—REVISED AND ENLARGED

CONTENTS

INTRODUCTION

It is rather difficult to classify and give proper definitions of the many varying kinds and types of dies used on the power press for rapid production of duplicate work. While there are, of course, some general classes into which all tools of this description may be divided, the various types overlap, so to say, and one is sometimes in doubt as to the proper classification of tools which combine the features of different types. In the following, however, the distinctions between the main types have been pointed out in general outlines, the definitions being broad enough to permit of adjustment according to special conditions.

All dies may, in the first place, be divided into two general classes: *cutting dies* and *shaping dies*. Cutting dies include all dies which simply cut or punch out required pieces of work from the stock fed into the press, without changing the condition or form of the stock in the plane in which it was located in the material from which it is cut. Shaping dies include all dies which change the form of the material from its original flat condition, producing objects in which the various surfaces are not in the same plane. The last mentioned main division often includes also the characteristics of the first; that is, some shaping dies are, for instance, a combination of cutting and shaping dies, the blank for the work to be shaped or formed being first cut out to the required outline from the stock, and then shaped to the desired form.

The main classes of dies, as will be recognized, are based on the *use* of the dies. The first of the classes mentioned, cutting dies, may, however, be further subdivided according to the *construction* of the various types of dies in this class. We then distinguish between four distinct types, *plain blanking dies*, *follow dies*, *gang dies*, and *compound dies*.

Plain blanking dies are the simplest of all types of dies, and are used to cut out plain, flat pieces of stock having, in general, no perforations, the work being turned out complete at one stroke of the press.

Follow dies, not infrequently also termed tandem dies, are used for work which must be cut out from the stock to required shape, and at the same time be provided with holes or perforations of any kind. The principle of the follow die is that while one part of the die punches the hole in the stock, another part punches out the work at a place where at a former stroke a hole has already been punched, so that a completed article results from each stroke of the press, but, in reality, two operations have been performed on the work before completion. The follow die cannot be depended upon to turn out very

accurate work, because it depends largely on the skill and care of the operator for the production of duplicate work. In both the plain blanking and the follow dies, the punch, or upper member, and the die, or lower member, of the complete tool, are distinct elements, the work being cut out or perforated by the entering of the punch into the holes provided for it in the die.

Gang dies are used when several blanks are punched out simultaneously from the stock. The advantage of the gang die over the plain blanking die is the saving of time.

Compound dies differ from plain blanking and gang dies in that the simple punch and die elements are not separated, one in the upper and one in the lower half of the complete tool, but these elements are combined so that both the upper and the lower part contain each a punch and a die. The faces of both punches, dies and strippers are normally held at the same level, and the strippers are spring supported so as to give way when the stock is inserted between the faces, and the press is in action. The springs are so adjusted that they are strong enough to overcome the cutting resistance of the stock, after which they will be compressed until the ram reaches the end of its stroke. A compound die produces more accurate work than the three types previously referred to, for the reason that all operations are carried out simultaneously at one stroke, while the stock is firmly held between the spring-supported opposing die faces. The disadvantage of the ordinary compound die is the difficulty encountered in "setting up," and the complexity of the design, which usually requires more or less frequent repairs.

The second main division of dies, the shaping dies, cannot be subdivided according to the construction of the dies in the same manner as the cutting dies. Shaping dies are usually designed more or less on the compound principle, outlined above, but owing to the great variety of work performed in these dies, the designs vary too greatly for a classification on the basis of constructional features. They may, however, be divided into sub-classes according to the general use to which they are put. We would then distinguish these four main subdivisions: *bending dies*, *forming dies*, *drawing dies*, and *curling dies*.

Bending dies are used when part of the surface of a piece of work is pushed from its original plane into a new shape in such a manner that the bent work does not form a closed curve.

Forming dies are used when the blank is required to be formed into a hollow shape, by being pushed into a cavity in the die.

Drawing dies are used for the same purpose as forming dies, but the process differs in that the outer portion of the flat blank to be formed is confined between two rigid flat surfaces, so that, when drawn radially inward from between them, no wrinkles can form.

Curling dies are used for bending over the ends or edges of the work into a circular cross section, like the turning over of the edges of hollow objects of sheet metal, etc.

Finally, we must mention the sub-press die, which, however, cannot be defined as a special class of die, but merely as a principle on which

all the different classes of dies, cutting as well as shaping dies, may be worked. The sub-press principle is simply that the upper and lower portion of the die, the punch and die, are combined into one unit by guide rods fastened into the lower part of the die and extending through holes in the upper part, or by some other provision for guiding. This construction permits of a high degree of accuracy eliminates the necessity of lining up the punch and the die each time they are set up on the press, and thus saves a great deal of time and cost.

In the following, we shall, however, deal only with the simpler forms of cutting dies, plain blanking and gang dies, except in Chapter VI, where reference will also be made to some of the more complicated types of dies.

CHAPTER I

METHOD OF MAKING BLANKING DIES*

From a mechanical standpoint it can truthfully be said that we are living in an age of dies. Never before has the industrial world made use of the punch and die as it is doing to-day. And no wonder; for this useful tool in all its different phases has proved beyond all reason-

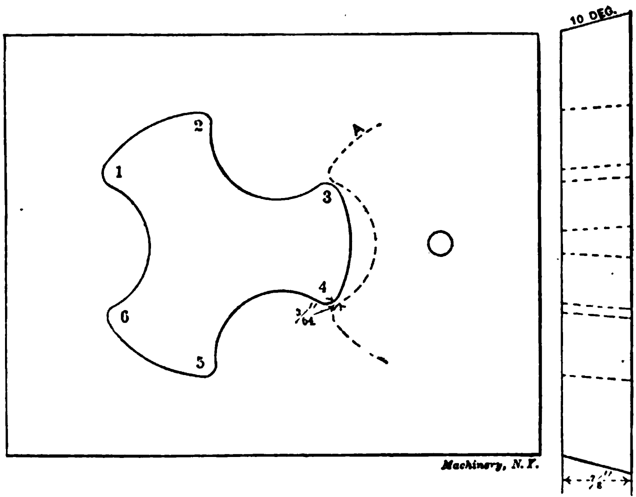


Fig. 1. Die used as Example in Illustrating Principles of Making Blanking Dies

able doubt that it can turn out more work in less time than the combined efforts of a room full of milling machines, shapers, and drill presses. To those who are unfamiliar with the die and its work the above may not appear feasible; but one has only to visit a modern

* MACHINERY, June, 1906.

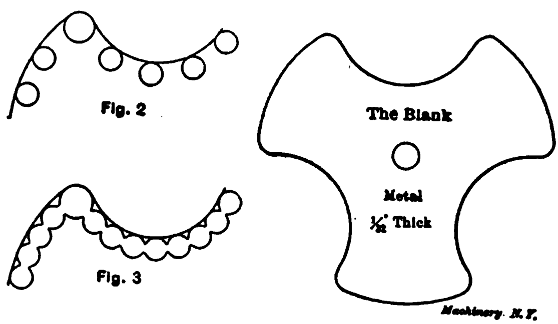
sheet metal factory to be convinced of the surprising rapidity with which the power press with its punches and dies will turn out not only work of all kinds of shapes and sizes, but accurate work as well.

Of the many different kinds of dies in use, the blanking die is probably the most widely employed. The reason for this is that almost all work that requires the use of various other kinds of dies has its beginning with the blanking die; for it is this die that cuts the work from the flat stock before it is completed by the other dies. In making the blanking die there are a few essential points to be taken into consideration, among which are the following:

1. Use good tool steel of a sufficient length, width, and thickness to enable the die to hold its own.

2. In laying out the die, care should be taken that as little of the stock as possible is left over, as waste, in cutting out the blanks.

3. Be sure not only that the die has the proper amount of clearance (which should be no more than two degrees and no less than one degree.) but also that the clearance is filed *straight*, so as to enable the blanks to readily drop through.



Figs. 2 and 3. Method of Removing Surplus Stock or "Core"

4. In working out the die, machine out as much as possible; don't let the file do it all.

5. In hardening the die, do not overheat it, as the cutting edge of a die that has been overheated will not stand up to the work, and requires so much sharpening in order to produce perfect blanks, that at its best it is nothing more than a nuisance.

In laying out the blanking die, the face of the die is first polished smooth and drawn to a blue color by heating. This gives better satisfaction by far than using coloring acid, for it gives a clear white line on a dark surface to work to, and is easier on the eyes, particularly when working by artificial light as is often necessary. When the die to be laid out is a blanking and piercing die, allowance of 3/64 inch must be made for the "bridge," *i. e.*, the narrow strip of metal that separates the holes in the stock from which blanks have already been cut. Fig. 1 shows how this is done; the dotted line *A* is drawn merely to show how the die is laid out.

After the die is laid out it is ready to be worked out. Now there

are several different ways of working out the surplus stock in a die of this kind. One is to drill say a half-inch hole at a safe distance from the line, and then fasten the die in a diemaker's milling machine and mill out the stock close to the line with a taper milling cutter, which gives the die the necessary clearance, thereby saving considerable time when filing out the die.

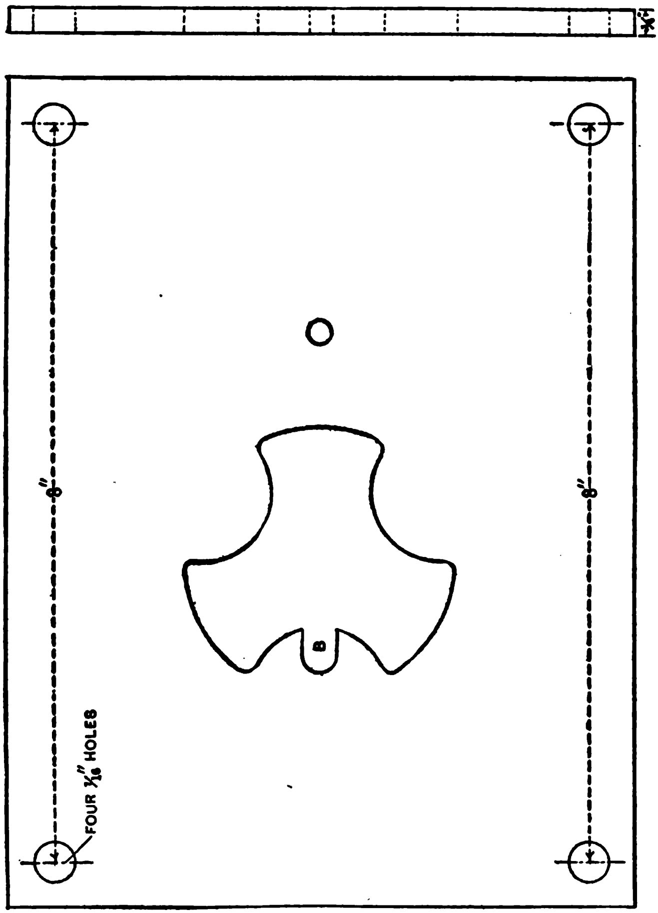


Fig. 4. Stripper Plate

Another method which is most commonly used, is to drill out the surplus stock on a drill press, after the manner shown in Figs. 2 and 3, which is done as follows: The six holes for the corners numbered 1, 2, 3, 4, 5, 6, Fig. 1, are first drilled and reamed taper, after which the other holes are drilled. These holes are drilled an even distance apart, and must therefore be spaced off, and then spotted with a prick punch before they are drilled. The best way to do this is to first

scribe an inside line at a distance from the outside line equal to one-half the diameter of the holes to be drilled, then space off, and spot. In spacing off, do not use dividers, but use a double prick punch. Using a pair of dividers requires too much time, besides the points get dull quickly enough without using them when it is unnecessary.

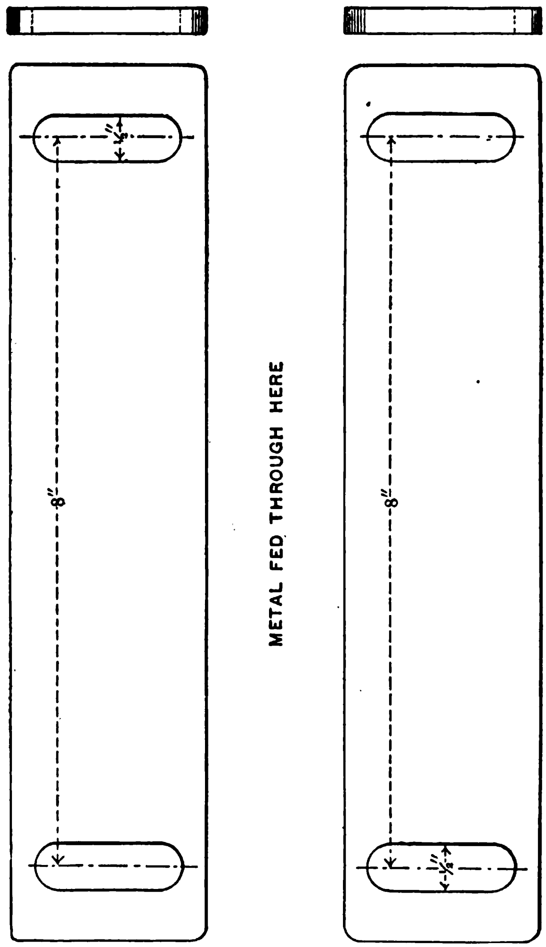


Fig. 5. Gage Plates

After the centers have been lightly spotted with the double prick punch, use an ordinary prick punch and make the spots a trifle deeper, so that the drill will more easily take hold.

In drilling, use the method shown in Figs. 2 and 3, for in this way the holes can be drilled closer together, thereby making it easier to get rid of the surplus stock and saving the time of broaching out the webs. The die blank should be slightly tipped by placing a narrow

strip of flat stock under the edge of same, as shown in Fig. 6, when the die is being drilled. This is done to give the necessary clearance, and does away with that time-killing operation of reaming the holes with a taper reamer from the back after they are drilled. After the surplus stock is gotten rid of, the die is finished by filing, using a coarse file to begin with, and finishing with a smooth one.

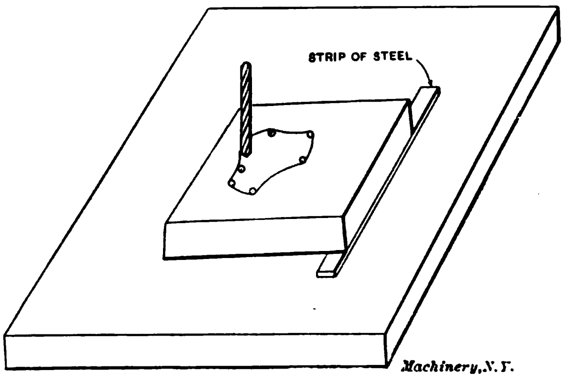


Fig. 6. Method of Obtaining Clearance when Drilling out the "Core"

Usually the die is made to fit a sample blank or a templet. This is done by entering the templet from the back as far as it will go after the die has been filed to the inside of the line. A lead pencil is then used to mark those parts of the die where the templet bears.

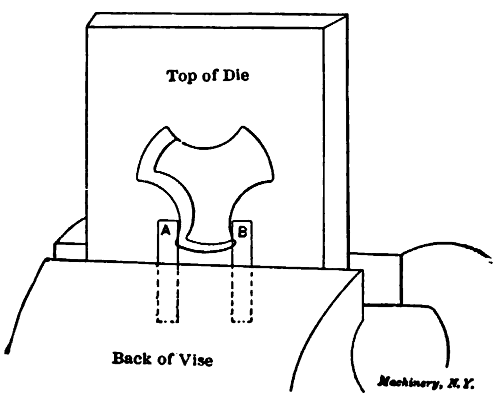


Fig. 7. Guarding the Corners in Filing the Die

The templet is then removed, the pencil marks filed out, the templet again entered and so on, until it is worked through the die. In filing out a die of this kind, where there is any danger of injuring that part of the die which has already been finished, use two strips of sheet steel, *A* and *B*, in the manner shown in Fig. 7, the round corners which are already finished being thus protected from the edges of the file.

In hardening the die, heat it to a cherry red, preferably in a gas furnace or a clean charcoal fire, and dip endwise into the solution used for hardening. When the die is sufficiently cold so that it can be taken hold of by the hands, withdraw it quickly and place it on the fire until it has become so warm that it will make water sizzle when dropped thereon; then immerse once more until cold. This is done to relieve the internal strains caused by hardening, and acts as a preventive to cracking. The face of the die is now polished, and the temper drawn to a light straw color, after which the die is allowed to cool of its own accord in oil. When cool, the die is ground on the top and bottom on a surface grinder, and if required it is lapped to size, which completes the operations.

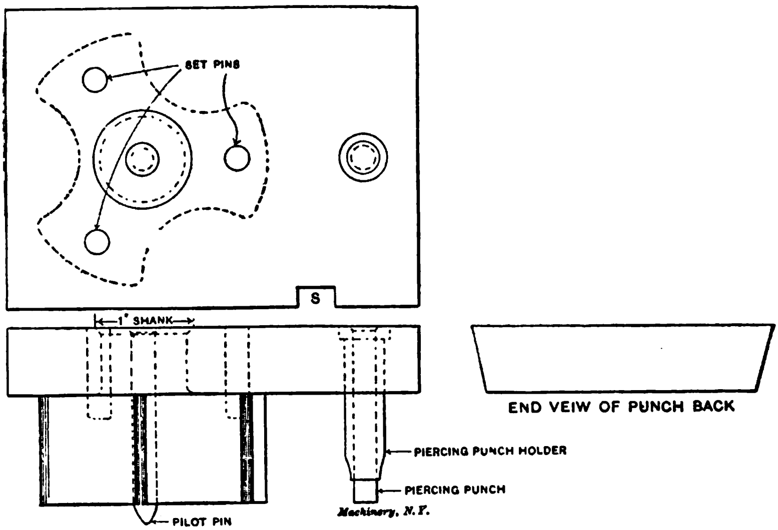


Fig. 8. Punch used with Die in Fig. 1

The punch is made after the manner shown in Fig. 8, and needs very little explanation. The dovetail punch back shown holds the punches in position, and is securely held in the press by the aid of a key. The slot *S* forms a position stop by engaging in a stud in the dovetail channel in the ram of the press, thereby eliminating the necessity of again resetting the tools in case the punch requires sharpening. The blanking punch is made from a tool steel forging, and is machined and sheared through the die in the usual manner. The one-inch shank is made a good driving fit in the punch back, and is upset as shown after the punch is driven in. The three set pins help to more securely hold the punch in position, and prevent it from turning.

The piercing punch is held in position by the piercing punch holder, which is driven tight in the punch back. The piercing punch is lightly driven in, and is made of drill rod, and can be very readily replaced

in case it is broken. The pilot pin is also made of drill rod, and can be very easily and quickly taken out when the punch requires sharpening.

The stripper and gage plates for this die are shown in Figs. 4 and 5. They are fastened by four 7/16 cap-screws to the die bed, used for holding the die in position when in use, and form, without doubt, not only the best, but by far the cheapest of the various methods employed for this purpose. While this method cannot be used on all kinds of blanking dies, it can, however, be used with the best of results on dies similar to the one described, and eliminates the unnecessary operation of drilling and tapping holes in the die itself to hold the stripper and gage plates in position. Not only that, but the gage plates as shown are used in connection with many other dies of a similar nature, thereby doing away with the necessity of having a set of gage plates for every die, as would otherwise be the case.

As the illustrations speak for themselves, no more explanation seems necessary, except perhaps that the slot *B* shown in Fig. 4 is to allow for an automatic finger to act as a position stop for the metal when it is run through.

CHAPTER II

BLANKING AND PIERCING DIES FOR WASHERS*

One of the simplest dies to make, coming under the head of blanking and piercing dies, is perhaps the die for blanking and piercing brass washers. The reason for this is that in making this die, the file and vise are not used; the construction and shape of this die are such as to allow it to be made by machinery. To lay out a single washer die is a very easy matter, but to lay out a die for cutting two or more washers at one time, so as to cut the greatest amount of blanks from the least amount of stock, is not understood as it should be. One of the reasons for this is that it is the custom in some shops to have the foreman, or some one else appointed by him, lay out all the dies before they are given to the diemaker to work out.

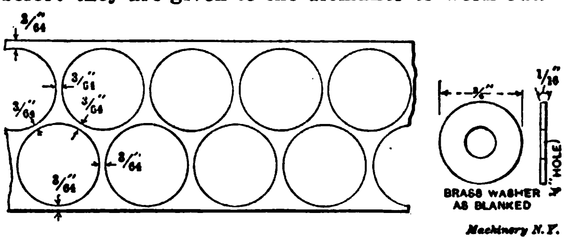


Fig. 9. Stock after having been run through the Die in Fig. 10, and Washer made

In laying out a washer die for blanking two or more washers at one time, one of the main points to be remembered is that all the holes from which the blanking and piercing are done must be laid out in an exact relation to each other, so as to eliminate the possibility of "running in" (*i. e.*, cutting imperfect, or half blanks, by cutting into that part of the metal from which blanks have already been cut). The required amount of blanks must also be considered, for it sometimes happens that the amount wanted does not warrant the making of a die that will cut more than one at a time.

Fig. 10 shows how a die is laid out for blanking and piercing two washers at one time, so as to utilize as much of the metal as possible. As shown, the ¾-inch holes marked *C* and *D* are the blanking part of the die, while the ¼-inch holes *A* and *B* are the piercing part. The distance between the center of *C* and *A* is 51/64 inch, as is also the distance between *D* and *B*. By referring to Fig. 9, which shows a section of the stock after it has been run through this die, it will be seen that there is a narrow margin of 3/64 inch of metal, known as "the bridge," between the holes. In laying out the die this margin must be taken into consideration, which is done in this manner: diameter of washer to be cut plus bridge equals distance from center

* MACHINERY, October, 1906.

to center, *viz.*, 3/4 + 3/64 = 51/64. The dotted circle shows that the die is laid out so that one washer is skipped in running the metal through at the start. This is done in order to make the die a substantial and strong one. It can be very readily seen that if the circle *E* were the blanking part instead of *D*, the die would be a frail one, and would not be strong enough for the work for which it is intended.

Another important point in laying out a die of this kind is to lay

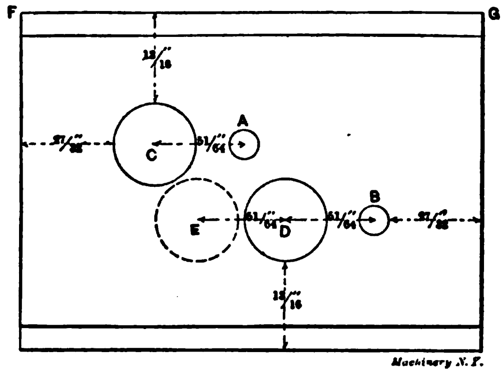


Fig. 10. Plan View of Die for Punching Two Washers Simultaneously

out the die "central," *i. e.*, laying out the die so that when it is keyed in position ready for use in the center of the die bed, it will not have to be shifted to the right or left side in order to make it line up with the punch. It may not be amiss to say in connection with the above that the punch back which holds the blanking and piercing punches in position should also be laid out "central"; this will be more fully described later on.

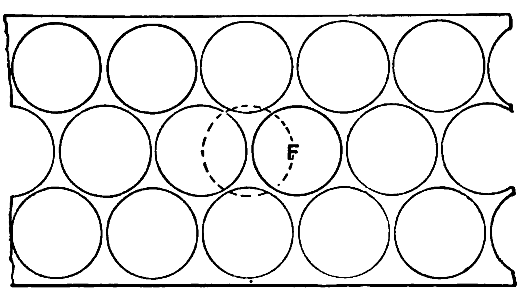


Fig. 11. Stock after having been run through Die in Fig. 12

Fig. 12 shows the layout for blanking and piercing three washers at one time, and hardly needs any explanation; the explanation given in connection with Fig. 10 sufficiently explains Fig. 12.

Fig. 11 shows a section of the stock after it has been run through this die. It can be seen that the holes match in very close together, and that very little stock is left. It is also seen that the three holes punched are not in a straight line, so far as the width of the metal is

concerned. This is done in order to save metal; the dotted circle *F* is merely drawn to show that wider metal would have to be used if the holes were in a straight line.

Fig. 13 shows the plan of a die for blanking and piercing eight washers at one time. The parts which are numbered are the blanking parts, while the parts that are lettered are the piercing parts of the die. This die is laid out similarly to Fig. 12, with the exception that there is provision for eight blanks instead of for three. Fig. 14 shows a section of stock after it has been run through this die. To give a better idea as to how the blanks are punched out in the manner shown, the sixteen holes in the metal from which blanks have been cut are numbered and lettered the same as the die. It should be understood

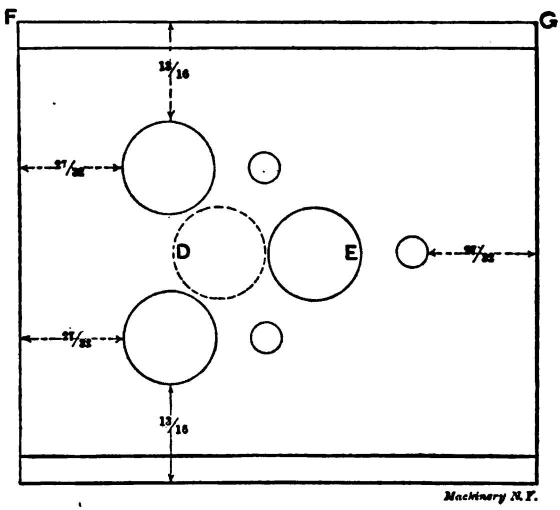


Fig. 12. Plan View of Die for Punching Three Washers Simultaneously

that the metal is fed through in the usual way, which is from right to left, and that the ¼-inch holes are first pierced out, before the ¾-inch blanks are cut.

By referring again to Fig. 13, the lay-out for cutting two, three, four five, six and seven blanks can be determined. The parts numbered and lettered 1—*A* and 5—*E* are the lay-out for two blanks. For three blanks: 1—*A*, 2—*B*, and 5—*E*. For four blanks: 1—*A*, 2—*B*, 5—*E*, and 6—*F*. For five blanks: 1—*A*, 2—*B*, 3—*C*, 5—*E*, and 6—*F*. For six blanks: 1—*A*, 2—*B*, 3—*C*, 5—*E*, 6—*F*, and 7—*G*. For seven blanks: 1—*A*, 2—*B*, 3—*C*, 4—*D*, 5—*E*, 6—*F*, and 7—*G*.

The die bed used for holding the die in Fig. 13 in position when in use should have its dovetail channel running in the direction *KL*, while the dovetail channel for the dies shown in Fig. 10 and 12 should run in the direction *FG*. The reason for this is that a longer bearing surface for the dovetail is obtainable by such an arrangement.

It should be remembered that all holes in dies of this kind are lapped or ground to size after hardening; they should be perfectly round and have 1 degree clearance. In some shops the holes are left straight for ¼ inch, and then tapered off 2 degrees.

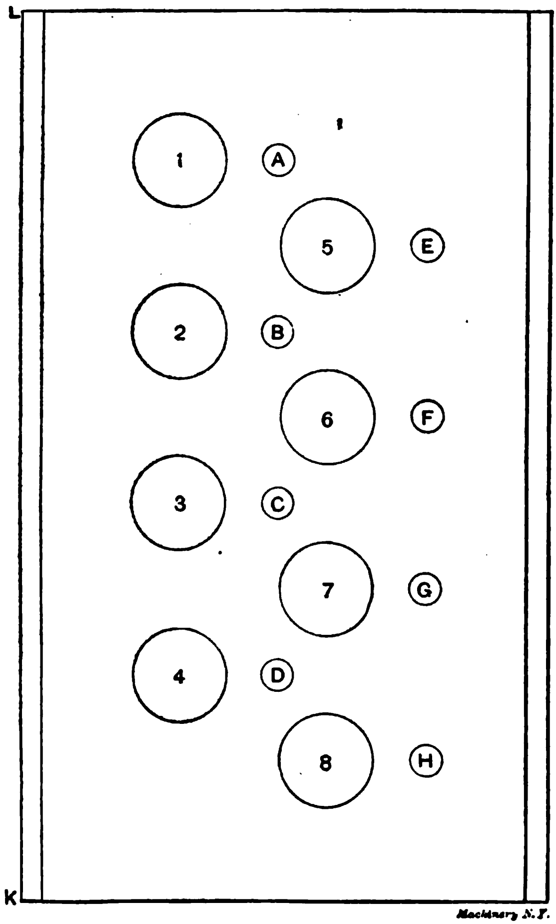


Fig. 13. Plan View of Die for Punching Eight Washers Simultaneously

An important point to bear in mind in making the punch is to have a perfect "line-up." It may not be generally known, but it is nevertheless a fact, that blanking tools that blank, or that pierce and blank two or more blanks at one time, will run longer without sharpening, cut cleaner blanks, and, in fact, give all around better results, if the punches are a perfect "line-up" with the die, than if they are lined up in the so-called "near enough" way. A perfect line-up, as referred to in the above, is a line-up that will allow a punch that consists of two

or more punches to enter the die the same as if the punch consisted of just one punch. The advantage of the perfect line-up over the other is that when in use the punches do not come in too close contact with the edges of the die. They enter the die, but do not bear against the edges in such a way as to dull the die, or round over the sharp cutting edge of the punch.

A punch that is almost a perfect line-up will enter the die, but it requires more force to make it enter. Why? Because in entering, one of the punches, for instance, rubs hard against the side of the die, and

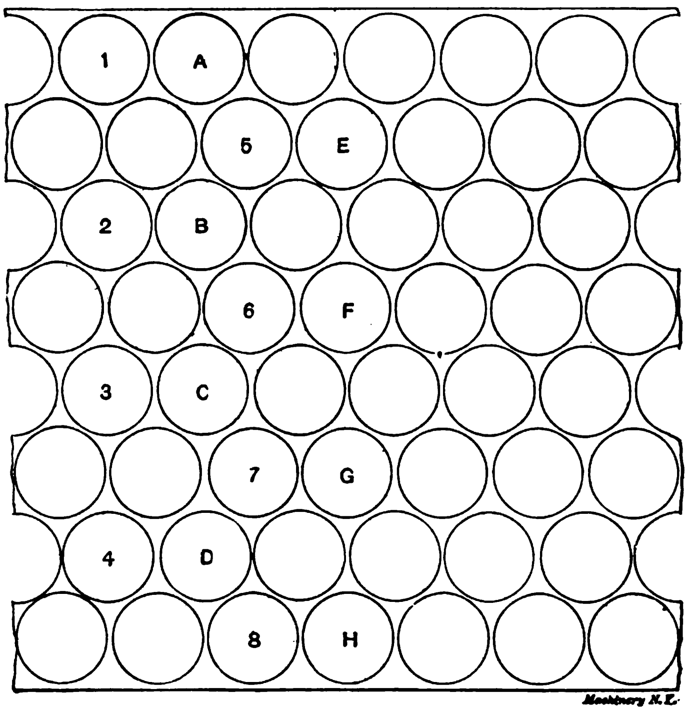


Fig. 14. Stock after having been run through the Die in Fig. 13

if set up in the press and allowed to run, that punch, no matter how small, will dull the edges of the die as well as the edges of the punch itself. The result is that the press must stand idle while the tools are being sharpened, and if the real cause of the trouble is not remedied, it is "the same old thing" over and over again.

Just a few words in regard to making the punch. In making the punch, care should be taken that it fits into the die not too loose, nor too tight. The blanking punches are hardened and ground to size. The taper shank is finished to size after hardening, so that when the punches are driven into the punch back they will stand straight and not lean to one side.

In laying out the dovetail punch back, first clamp the back central on the face of the die. This is done so that when the punches are driven in position in the punch back, and set central in the ram of the press, ready to be used, no shifting is required in order to make the punch line up with the die, which is keyed in the center of the die bed. After clamping the punch back in this position, the blanking part of the die nearest the end is scribed on the face of the punch back. Do not scribe all the holes and rely upon finding the center of each circle thus scribed with a pair of dividers, and then true up these centers on a faceplate in order to get a perfect line-up; this method increases the chances of error, especially when there are six or eight

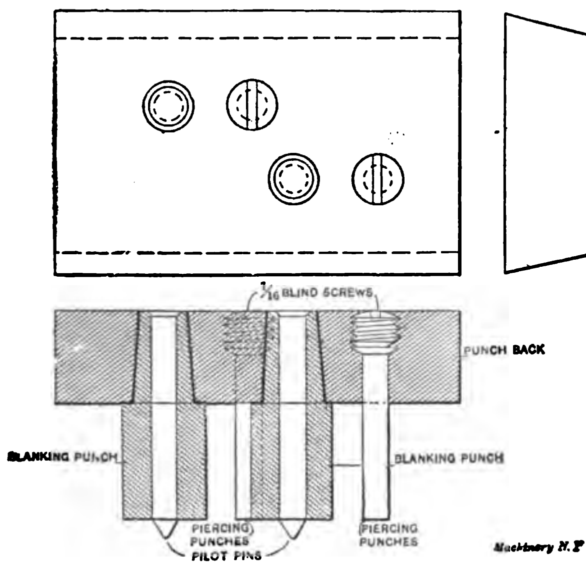


Fig. 15. Punch Back, with Punches Inserted

punches to be set in position. A better way is to scribe one circle as stated above, and remove the punch back from the face of the die; find the center of the circle scribed; true up this center, and drill and bore out the hole to fit the taper shank of the blanking punch.

Fig. 15 shows how a punch of this kind is made. The punch as shown is used with the die shown in Fig. 10. After the hole is bored to size, the already finished blanking punch is driven in tight in the manner shown. Two narrow parallels, say 1/2 × 3/4 inch, are now laid on the face of the punch back, and the blanking part of the die that corresponds with the punch driven in is slipped over the same, until the face of the blanking die rests upon these parallels, after which the die is clamped tightly thereon. The next hole is now trued up with a test indicator until the hole runs dead true. The die is then removed, the hole for the taper shank is worked out, and the

punch driven in. Where there are more punches to be set in, the same method is used until they are all in position. This insures a perfect line up, providing that ordinary care and precaution has been used in doing the work. In boring out these holes it is best to use a bolster having a dovetail channel, and to hold the punch back in position with a key. This is better than using straps to fasten the punch back to the faceplate, as the straps are likely to interfere with the parallels and the die, when locating the exact position for the holes to be bored.

In locating the position for the piercing punches, it sometimes happens that the holes are so small that they cannot be bored. The holes are then transferred by a drill that runs true and is the same size as the holes in the piercing die, the die being used, so to speak, as a drill jig.

Fig. 15 shows how the piercing punches are held in position. The punches are made of drill rod, and are prevented from pushing back by hardened blind screws as shown. If thin, soft metal is used, the method for holding the two pilot pins in position shown in the previous chapter may be employed. When the piercing punches are made and held in position as shown in Fig. 15, a spring stripper is sometimes used, and is fastened to the punch back, and the holes for the piercing punches in this stripper are made a sliding fit, in order to prevent the punches from springing or shearing. When the ordinary form of stripper is used, the piercing holes are also made a good sliding fit.

CHAPTER III

MAKING BLANKING DIES TO CUT STOCK ECONOMICALLY*

A most important point for the diemaker to bear in mind in making blanking dies for odd shapes is to lay them out so that the minimum amount of metal will be converted into scrap. In fact, hardly too much stress can be laid upon this one point alone. It is an easy matter to waste a considerable percentage of the stock by lay-outs which may *appear* to be fairly economical. The diemaker should make a careful study of the most economical relation of blanking cuts to one another and to the stock. It is the object of the present chapter to point out by actual examples how stock can be saved which may be converted into scrap if the diemaker is not constantly watching out for possible economies. As an illustration, it sometimes happens that by laying out the dies so that the blanks are cut from the strip at an angle of 45 degrees, as shown in Fig. 17, a considerable economy of metal can be effected over a right-angle arrangement, that is, one in which the dies are set so as to cut the blanks straight across the strip. The angular location permits the use of narrower stock and

* MACHINERY, February, 1907.

materially reduces the amount of scrap metal. Fig. 16 shows the plan of the die, and needs little or no explanation, as the manner in which it is laid out is obvious; the plan of the strip shown in Fig. 17 also clearly shows how the die is laid out.

Another method that is often used to save metal is that shown in Figs. 19 and 20. This method is used where the required amount of

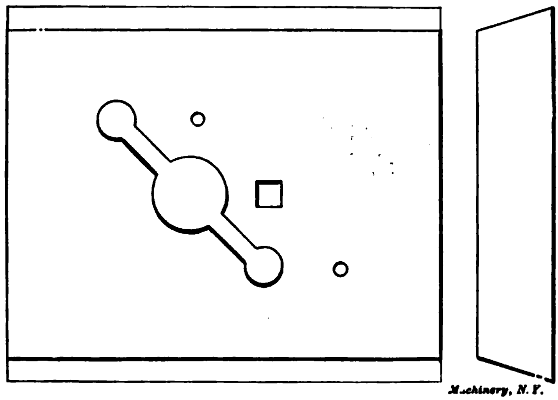


Fig. 16. Example of Blanking Die

blanks does not warrant the making of a double blanking die; also when, unavoidably, there is a considerable amount of stock between the blanks after the strip has been run through as shown at *A* in Fig. 19. To save this metal the strip is again run through in a reverse order after the manner shown in Fig. 20, thereby using up as much of the metal as it is possible to do. Besides blanking and piercing

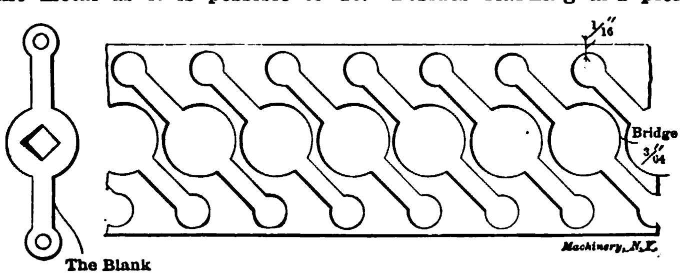


Fig. 17. Section of Stock after having been run through Die in Fig. 16

the blank when running the metal through the first time, the holes numbered 4, 5, and 6, Fig. 18, are also pierced. This is done for the reason that when the metal is run through the second time it prevents cutting of "half blanks" by "running in," or, in other words, the liability of cutting imperfect blanks by cutting into that part of the metal from which blanks have already been cut. This guiding action is effected by three pilot pins in the blanking punch (not shown) which engage the three pierced holes made when the strip was

run through the first time. The pilot pins engaging with the pierced holes cause the second lot of blanks to be cut centrally with the holes, and also to be accurately centered between the portions of stock from which the blanks have already been cut. When this die is in

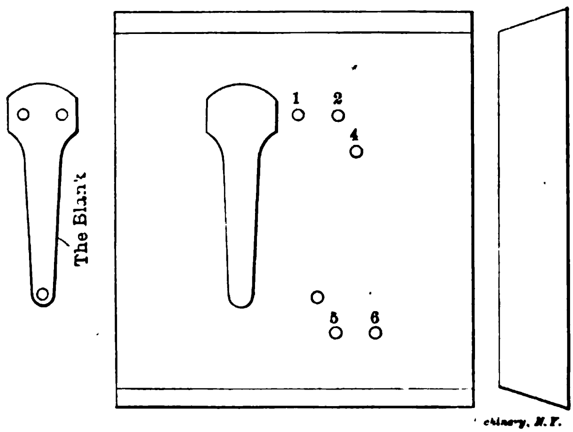


Fig. 18. Another Example of Blanking Die

use, the metal is run through in the usual way from right to left until half of the required amount of blanks is cut, after which the piercing punches for the holes are taken out and the metal is run through again and the other half of the required amount of blanks is cut.

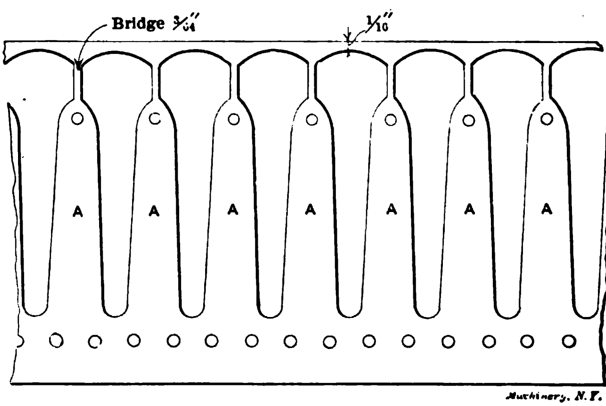


Fig. 19. Stock after having been run once through Die in Fig. 18

In laying out this die, which is done after the manner shown in Fig. 28, the line *A* is used as the center line for the piercing holes numbered 1 and 2 in Fig. 18, and the line *B* is the center line of the blanking part of the die. The line *C* is the center line that shows the center

of the next blank to be cut and is laid out 53/64 inch from the line *B*. This dimension is fixed by the fact that the widest part of the blank is 25/32 inch, and the bridge between the blanks is 3/64 inch, the sum of which equals the distance from center to center of adjacent blanks. The line *D* is the center line for the blank *C*, Fig. 20, which is cut when the metal is run through the second time, and is made at 0.414 inch or one-half of 53/64 from the line *C*, Fig. 28, inasmuch as the blank is cut centrally between that part of the metal from which the blanks *A* and *B*, Fig. 20, are cut.

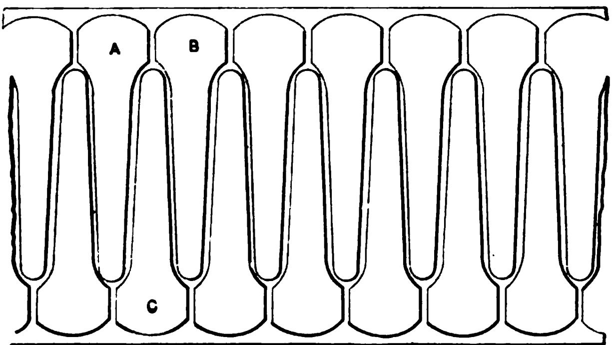


Fig. 20. Stock after having been run twice through Die in Fig. 18

Fig. 21 shows a double die for blanking and piercing brass, producing the shape shown in the sketch at the left; it is laid out so as to save as much of the metal as is practically possible without added expense so far as the operation of blanking and piercing is concerned. By referring to Figs. 22 and 23 it can be seen that the strip of metal from which the blanks are cut is run through a second time for reasons that will be given. One reason is that wider metal can

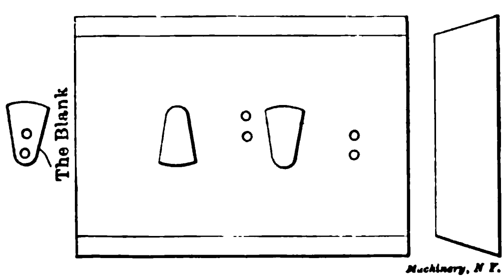


Fig. 21. A Third Example of Blanking Die

be used by doing this, which in itself is a saving so far as the cost of metal is concerned. Wide brass can be bought at a lower price per pound than narrow brass; the other reason is that a strip of metal 1/16 inch wide and as long as the entire length of the strip is saved

on every strip that is run through. If narrow metal were used, there would be waste of ⅛ inch of metal (*i. e.*, 1/16 inch on each side) of every strip run through, and on two strips from which no more blanks could be cut than from the wider strip shown in Fig. 23, there

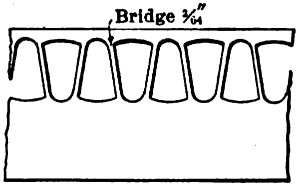


Fig. 22. Stock after having been run through Die in Fig. 21 once

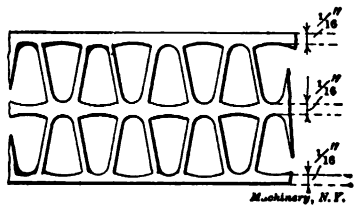


Fig. 23. Stock having twice been run through Die in Fig. 21

would be a waste of ¼ inch of metal. On the other hand, by using wide metal the waste would be only 3/16 inch, as indicated in the cut. Fig. 29 shows how this die is laid out, and should be sufficiently clear to explain itself.

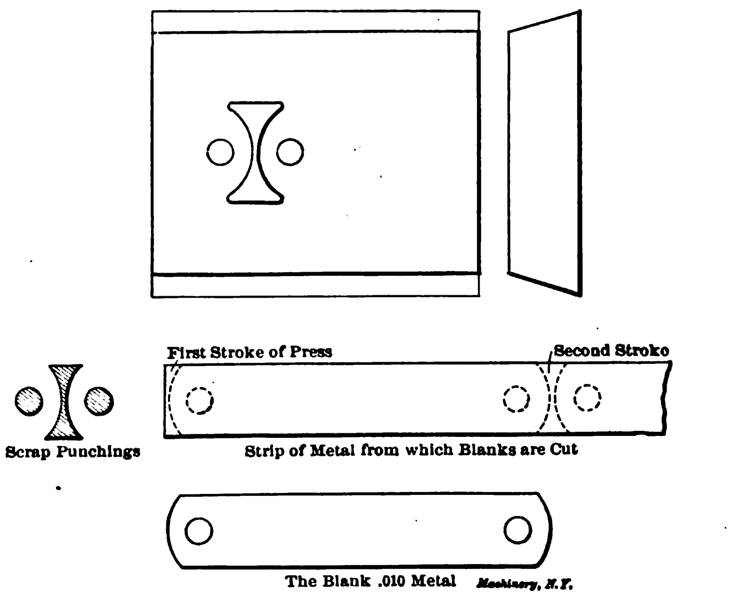


Fig. 24. Blanking Die for Producing Links

To fully understand the manner in which the metal is gradually worked up after each stroke of the press, short sections are shown in Fig. 25. At the first stroke four holes are pierced and two plain blanks —with no holes—*A A* are cut out. At the second stroke there are also four holes pierced and the two blanks *B B*, for which the holes were pierced at the previous stroke, are cut. At the third and fourth strokes

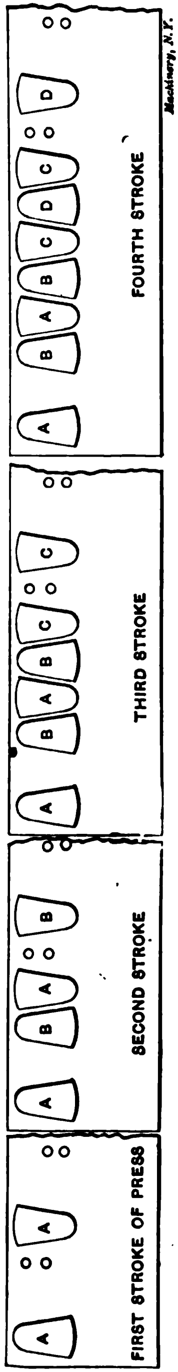


Fig. 25. Appearance of Stock after each Successive Stroke of the Press

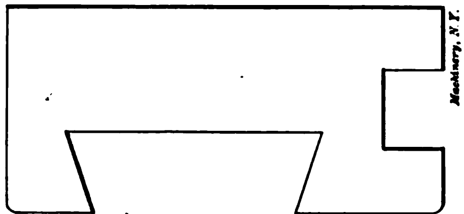


Fig. 27. Gage for Planing Die Blanks

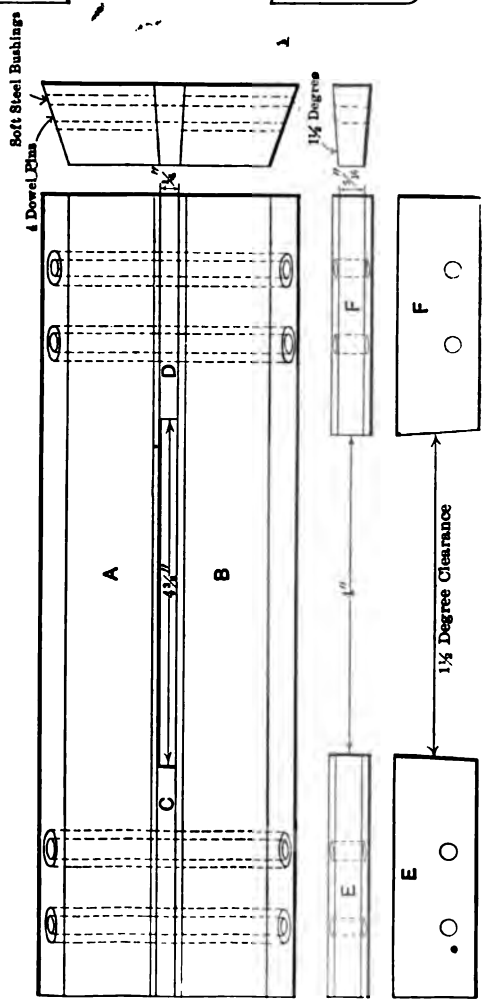


Fig. 26. Die with Interchangeable Parts, Permitting Two Sizes of Blanks to be Punched by Changing the Center Pieces only

the holes begin to match in with each other, as shown, so that when the metal is run through it will look like the strip shown in Fig 22. It should be borne in mind that four holes are pierced and two blanks are cut at each stroke of the press; also that the metal is fed after each stroke a distance equal to the distance from the center of *A* to the center of *B*, as indicated in the strip marked "second stroke," Fig. 25, and which is 25/32 inch (see Fig. 29). By way of further explanation it may not be amiss to state that the distances from the center of *A* to *B*, *B* to *C*, *C* to *D*, and *D* to *C*, as shown in the strip marked "fourth stroke" are each 25/64 inch, or half of 25/32 inch.

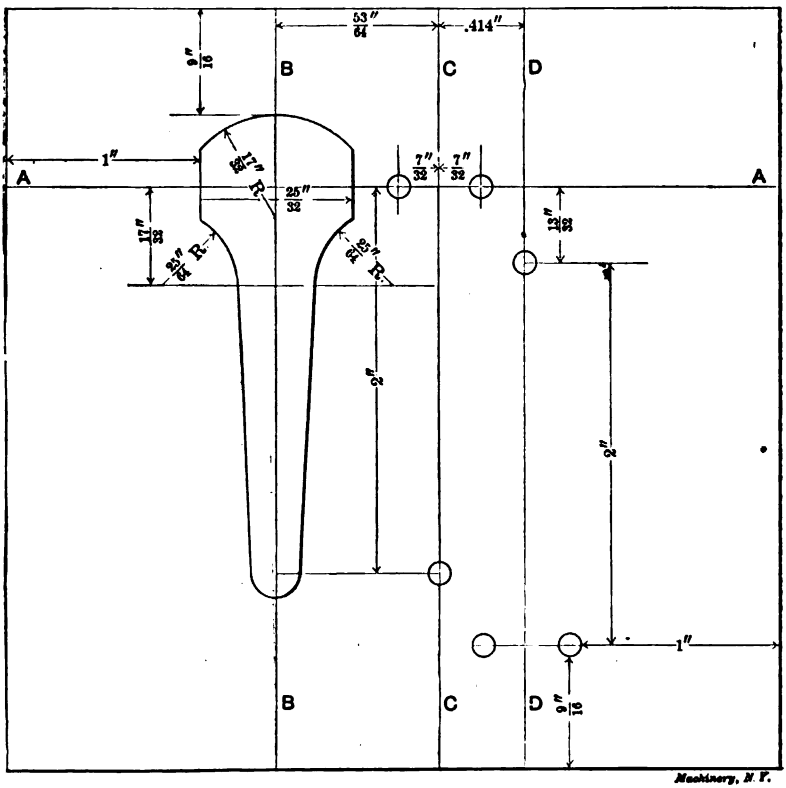


Fig. 28. Layout of Die Shown in Fig. 18

While the dies shown in Figs. 24 and 30 are commonly known, it may not be out of place to say a few words with reference to them, as they form an important part in the economical production of sheet metal goods. The first, or Fig. 24, shows a die that is used to produce, from narrow ribbon, a long blank with rounded ends and with a hole pierced in each end. The principal feature of this style of die is that there is very little waste of material in the production of the

blanks, as will be noted from the sketch of the scrap punchings shown at the left, and another feature is that by the aid of an adjustable stop, not shown, almost any length of blank can be made without alter-

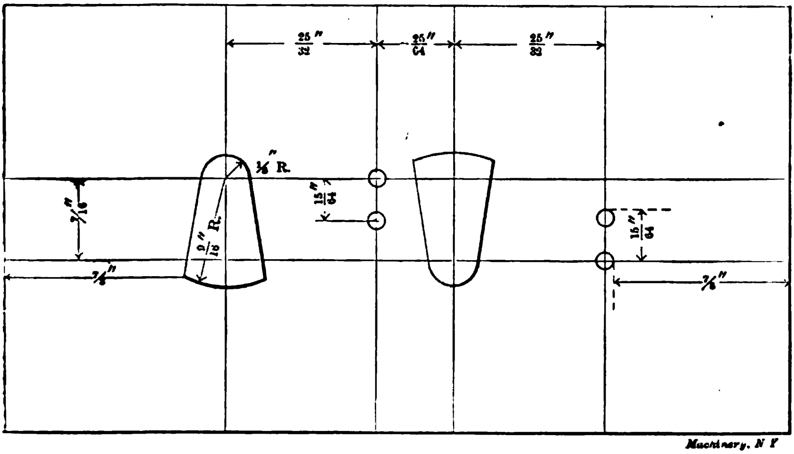


Fig. 29. Layout of Die Shown in Fig. 21

ing or resetting the tools after they have been set up in the press. The working part of the die is laid out a little to the left of the center so as to give sufficient length for the gage plates which are fastened to

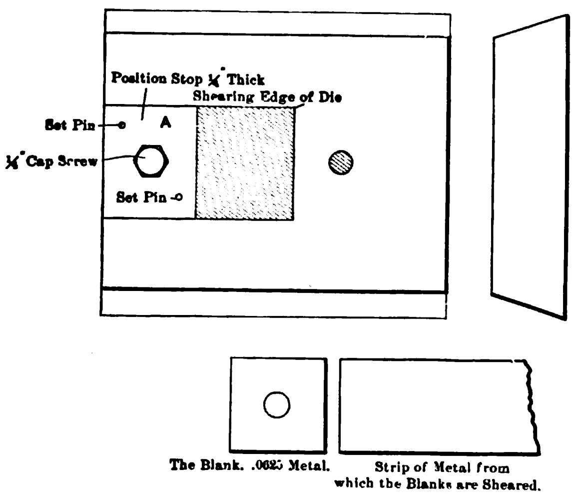


Fig. 30. Blanking Die for Square Washers. Shaded Portions in Die indicate Parts punched out from Stock

the die by ¼-inch cap-screws. These gage plates are used to keep the metal in position while it is being fed from right to left as the blanks are cut from the strip.

Fig. 30 is a combination piercing and shearing die and is used for producing the 1-inch square washer shown in the cut. The principal feature of this die is that there is no waste of metal in producing the blank, except, of course, the ¼-inch round punching taken from the center. The strip of metal in this case can be fed from right to left or front to back, as preferred.

CHAPTER IV

CONSTRUCTION OF SPLIT DIES*

A die of great importance in the production of sheet metal parts is the split die. There are two principal reasons for using the split die. One is that it sometimes happens that the blanks to be cut are of such a shape that the die can be more quickly and cheaply made by making a split die than by making a solid or one-piece die. The other reason is that when the required blank must be of accurate dimensions, and there is a chance of the solid die warping out of shape in hardening, the split die is preferred because it can be much more easily ground or lapped to shape.

Fig. 31 shows the manner in which the ordinary split die is usually made. After the die is worked out, it is hardened and ground on the top and bottom. The two sides *A* are then ground at right angles with the bottom.

The cutting parts of the die, *B*, are next ground at an angle of 1¼ degree with the bottom, so as to give the necessary clearance in order that the blanks may readily drop through. The key *D* is now set in place, and the die is keyed in the die bed by the aid of a taper key. The key *D* prevents the die from shifting endwise; the keyway should have rounded corners as shown, which not only give added strength, but also act as a preventive to cracking in hardening. The last operation is to grind the two circular holes. This is done by first lightly driving two pieces of brass or steel rod into the holes until they are flush with the face of the die. The exact centers are then laid out and spotted with a prick punch, care being taken so as to get the centers central with the sides *B*. The die is now fastened to the face-plate of a universal grinder, and the center mark is trued up with a test indicator until it runs exactly true. The brass rod piece is then driven out, and the hole ground to size, with 1½ degree taper for clearance. The other hole is next ground out in a similar manner, which completes the operations so far as the die is concerned. It often happens with a die of this kind that when it is placed in the die bed and the key driven in place, it will "close in." To overcome this, the die is relieved after the manner shown at *C*, which does not in any way prevent it from being securely held in place when in use.

* MACHINERY, March, 1907.

Fig. 26 shows a rather novel form of split die; this die with a slight change practically takes the place of two dies. It is used for piercing slots in brass plates. The size of the slot for one style of plate is 4⅜ inches long by ¼ inch wide; for the other plate the slot is 4 inches long by 5/16 inch wide. The cutting part of the die, shown in Fig. 26, is made in four sections, *A*, *B*, *C*, *D*. The cut fully explains itself and therefore needs no detailed explanation. It may not be out of place, however, to say that the soft steel bushings, as shown, are used to allow for the contortion of the parts *A* and *B* in hardening. It may be added that the four bushings shown in the piece *A* were driven in first; then solid pieces were driven in the part *B;* then the holes were drilled in these latter pieces, being transferred from the bushings in the part *A*. In Fig. 26 are also shown the parts used in connection with this die for piercing the 4 x 5/16-inch slot. These parts are made as shown, and are hardened only at the cutting ends. Outside of the fact that this style of die practically takes the place of

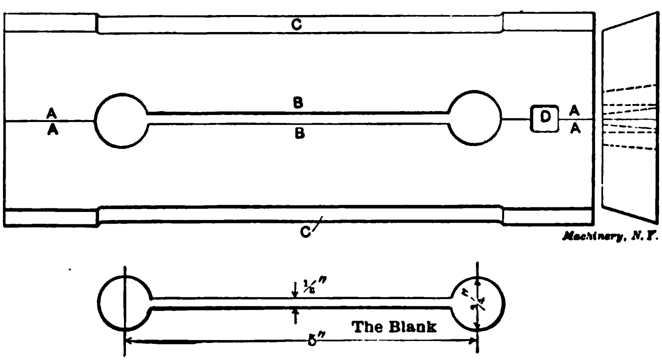


Fig. 31. Example of Split Die

two dies, there is still another feature in connection with it that will bear mentioning; there is no special or extra die bed required for this die when in use.

It may not be amiss at this time to say a few words with reference to die beds. (In some shops this part is called bolster, die block or die holder.) Perhaps the most commonly used and the best die bed for general use in the press room is the style of bed shown in Fig. 32. This die bed is principally used for the reason that the screws that fasten the die bed to the bed of the press do not have to be screwed entirely out, either in placing the die bed in the press or in taking it out, as the slots *C* and *D* are made at right angles with each other for just this reason.

The dovetail channel is planed so that when the die is keyed in position the center of the die is central with the slot *C*. The side of the die bed marked *A* is planed at an angle of 10 degrees, and is parallel with the slot *C*. The side marked *B* is planed at an angle of 13 degrees and is at an angle of 1 degree with the center line. The rea-

son for planing this side to an angle of 13 degrees instead of 10 is that the increased angle causes the die to lie flat, and prevents it from raising or tilting up in any way when the key is driven in.

In speaking of the key, it may well be added here that the taper-key method of holding blanking dies in the die bed is the best of the various methods which are generally used. The set-screw method is doubtless the poorest of all. The key as shown in Fig. 32 is driven in on the front side of the die bed. This is optional, however, as the practice differs. In some shops the key is driven in on the front side while in others it is driven in on the back.

Of late years there has been a tendency among large concerns to have all their die beds for the power press made from semi-steel castings, or of machine steel for certain classes of heavy work, instead of from gray iron as heretofore. This is being done because a gray

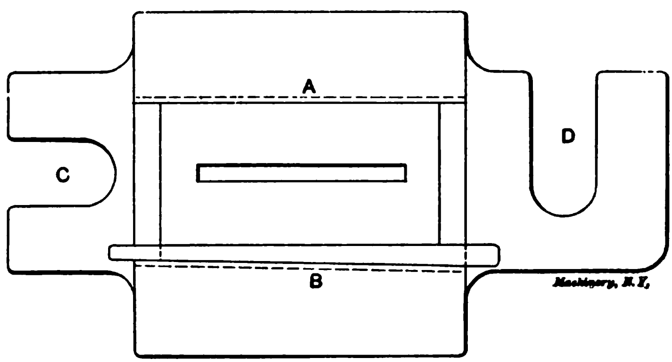


Fig. 32. Example of Die Bed

iron bed that is used day after day for holding dies for cutting heavy metal will not stand up during long and hard usage as it should. Past experience has proven that gray iron die beds in time become out of square; then, again, they sometimes crack. With the semi-steel, or the soft steel die bed, this does not happen. It has been found that semi-steel and machine steel die beds pay for themselves many times over.

In planing up the stock from which the blanking dies are sawed off before they are worked out, a gage similar to the one shown in Fig. 27 should be used for planing up the different widths of dies. In this way the dies will be of a uniform width and thickness, which makes it possible to have them interchangeable with the respective die beds for which they are used.

CHAPTER V

STOP-PINS FOR PRESS-WORK*

The stop-pin occupies a position of much importance among the accessories of the blanking die. Upon its design and adjustment depend both the quality and the quantity of the output of the press. Hence it is fitting that some attention be given to the consideration of it. By proper selection from the types to be described it is possible to secure a large output of blanks without recourse to more expensive apparatus. The several forms of stop-pins enumerated in the following list will be described in order, their proper uses being noted, together with their merits and faults: The plain fixed stop-pin; the bridge stop-pin; the simple latch; the spring toe latch; the side swing latch; the positive heel and toe latch; the gang starting device.

These devices are capable of giving, under the proper conditions, the maximum output of blanks. With the exception of the first, they can be used with either hand feed or automatic roll feed.

The ideal output of one blank for every stroke the press can make in a day is never realized, with single dies. The delays which arise

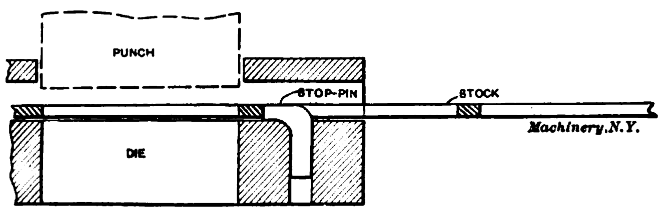


Fig. 33. A Plain Fixed Stop-pin

from so many sources have to be studied carefully and eliminated so far as they contribute to unnecessary expense. In addition to improper design and poor adjustment of the stop-pin, other causes of small output are: Lack of skill; inconvenient arrangement of the new stock, the blanks and the scrap; inefficient methods of oiling the stock; and poorly made or poorly designed dies. A skillful operator, if given a little freedom, will usually arrange the stock distribution quite well, but the design and adjustment of the dies and the stop-pin usually devolve upon the toolmaker.

Plain Fixed Stop-pin

The plain fixed stop-pin, which is the simplest form, is indicated in Fig. 33. With it the operators become so expert that they are able for several minutes at a time to utilize every stroke of a press making 150 revolutions per minute. This stop is best suited to the use of strip stock in simple dies, because a miss will then cause no serious

* MACHINERY, September, 1909.

delay. The time between finishing one strip and starting the next affords the necessary rest for the operator. The concentration required is very intense—especially for the novice. When but a few blanks are made from a die at one time, and when changes of dies are frequent, this simple stop-pin is the most economical. Of course, it would not be feasible to use this stop-pin for coiled stock and expect the operator to finish the coil without a rest or a miss. There is, however, one method of using this stop which permits of a maximum output: that is to allow no metal between the blanks. Then the stop-pin will ex-

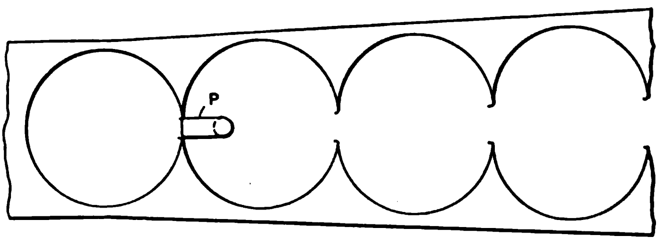


Fig. 34. Fixed Stop set close up to the Die so that there is no Stock between the Blanks

tend clear up to the die and be high enough so that the stock cannot jump it. Each blank will then part the scrap at the stop-pin and allow the stock to be pulled along to its next position. This arrangement is shown in Fig. 34, with the stock parting at the pin *P*. This method is widely used on simple work where the edge of the blank does not have to be perfectly uniform. Where the die has least to cut it will wear away most on account of the thin pieces of stock that crowd down between the punch and the die. Small drawn cups are

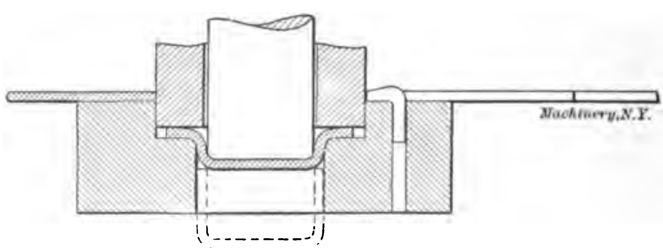


Fig. 35. Example of Work to which the Stop shown in Fig. 34 is adapted

made in this way. The blank is cut by the first punch and held by it while a second punch, within the first, draws the blank through another die and forms the cup. This is shown in Fig. 35. The stock feeds to the right and each cup, as formed, pushes the one ahead of it through the die as indicated by the dotted lines.

Bridge Stop-pin

The bridge stop-pin, shown in Fig. 36, is perhaps the most efficient and easiest to operate of all. It is also the simplest in design. The stop-pin *P* projects downward from a bridge *B* that extends over the

stock which is being fed to the left. Provision is made for the blank (or scrap, as the case may be) to fall out under the bridge. Its use is limited, however, to that class of work which cuts the stock clear across and uses its edges as part of the finished blank. As here shown, the scrap is being punched through the die, and the blank when cut falls down the inclined surface shown. When the blanks are simpler and have straight ends, the die may be so arranged that each stroke finishes two blanks, one being punched through the die and the other

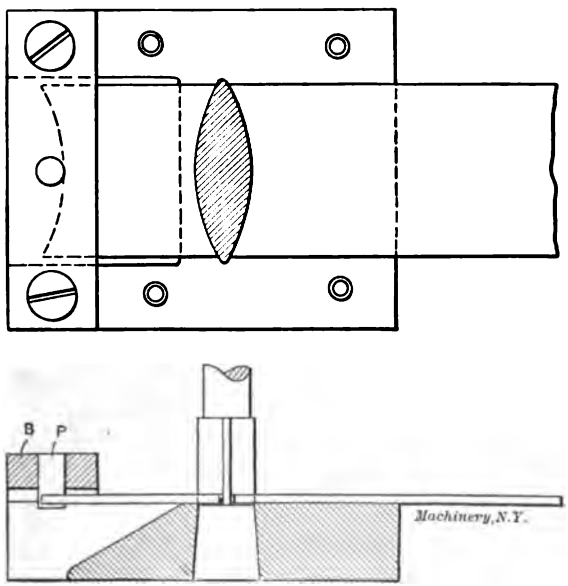


Fig. 36. The Bridge Stop-pin

falling outside down the incline. Little skill is required of the operator; he simply has to be sure to push the stock up to the stop-pin at each stroke.

The Simple Latch

The simple latch is shown in Fig. 37. It is suited for dies that have pilot-pins. The latch is lifted by the down stroke of the punch and is lowered again as the punch rises. Hence it is evident that, if used with dies without pilot-pins, the punch must reach the stock and hold it before the latch lifts. When its lifting is thus delayed it will lower before the punch withdraws from the stock and will fall in the same place it lifted from. The stock will then not be fed along. But if a pilot-pin is used, it may be set so as to enter the guide hole just before the latch lifts. The latch may be set to lift before the punch reaches the stock. It will then fall after the punch withdraws from the stock, and sufficient time may be allowed for the operator to feed the stock along. This device is best suited for use with automatic feed rollers because the timing of the operations would be more uniform; whereas

if the operator does not pull the stock with uniform speed the latch is apt to drop too soon or too late. Another manner of operating this simple latch is to give it its motion by means of a cam or eccentric on the press shaft. When thus driven its motion can be very care-

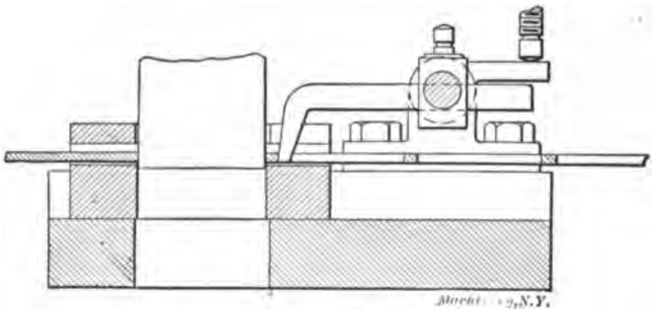


Fig. 37. The Simple Latch Form of Stop

fully timed, irrespective of pilot-pins. This style is also best suited for automatic roll feed. New presses are often provided with this attachment.

The Spring Toe Latch

The spring toe latch involves but little change from the simple latch. Fig. 38 shows it clearly with an enlarged detail of the spring

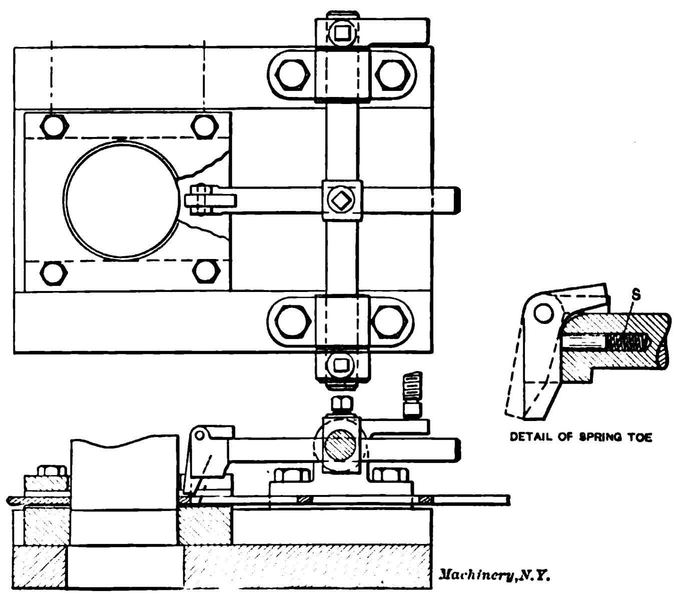


Fig. 38. The Spring Toe Latch Stop

toe. This latch may be used very successfully with hand feed and there is little danger of the stock getting by it too fast. Its operation

is as follows: As the punch lowers and starts to cut the blank, an adjustable screw on the ram or punch plate lifts the latch. Its spring toe snaps forward and when the latch lowers, it rests on the scrap left between two blanks; hence it cannot fall back into its former place. When the operator pulls the stock along, the latch toe drops into the next hole and brings the stock to a stop at the proper point, compressing the light spring *S* as it does so. This design is simple, rigid and effective. The spring toe here shown is preferable to the design which follows because it is light and requires but little tension on the stock to bring it to a stop.

The Side Swing Latch

The side swing latch is shown in Fig. 39 and is but a modification of the latch shown in Fig. 38. When the punch descends, an adjustable screw hits lever *L* and lifts the latch. The whole rod *R* then springs forward till collar *C* stops against *B*. When the latch lowers it rests on the stock as did the spring toe latch. As the stock is pulled along, the latch drops into the next hole and acts as a stop

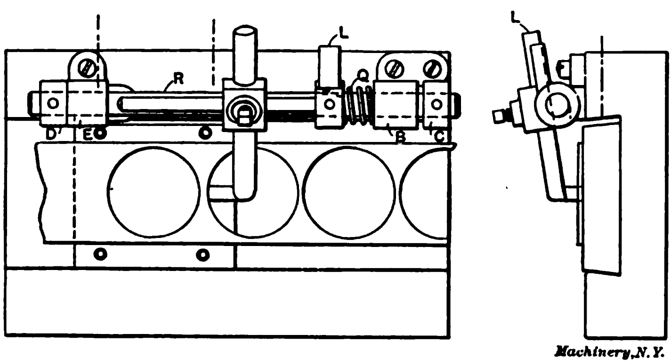


Fig. 39. The Side Swing Latch Stop

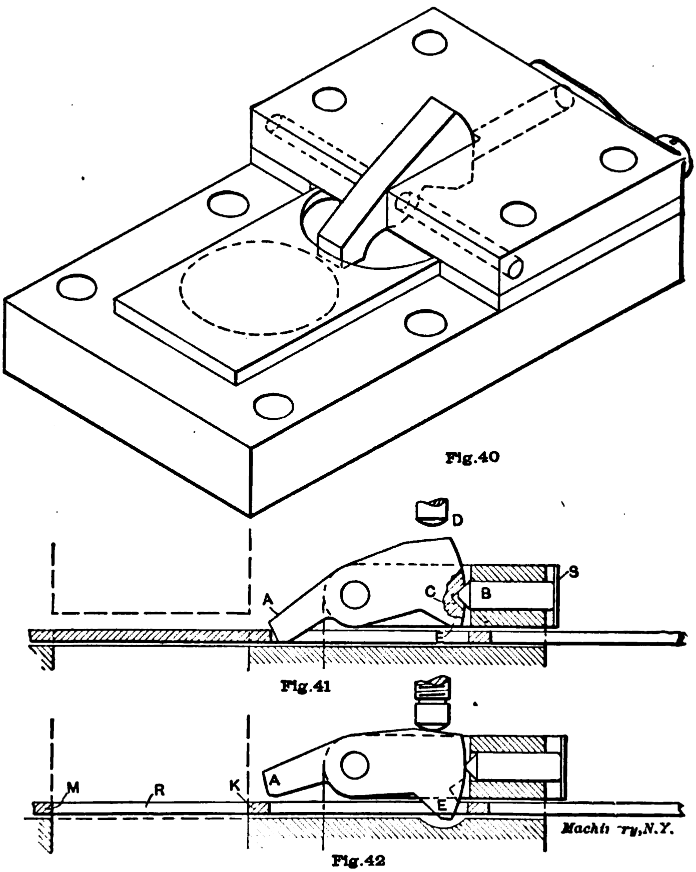
again. In this style the tension on the stock must be greater than with the spring toe latch, because the whole rod *R* has to be pulled along against the spring *Q* until collar *D* stops against *E*. If this design were modified, however, so that the side bearings would be used only for allowing the latch to swing, the toe could be constructed like the spring toe latch and would then be quite as effective as this type, though not so rigid.

Positive Heel and Toe Latch

While the two previous automatic stop-pins rely on gravity or a spring to bring them back in position, the heel and toe latch is positively operated. It is shown in Fig. 40, with the stripper removed. Its distinctive feature, which recommends it for use on a large variety of work, is that it is impossible for the stock to slip by it faster than one blank per stroke of the press. This is a very important matter when combination or gang dies are being used, because the pilot-pins so widely used require the guide holes to be punched just ahead of them. If the stock slips too far, the guide holes pass be-

yond the pilot-pins, and when the punch descends, the pilots punch their own holes, throw down a heavy burr and cause a delay—if nothing more serious.

Fig. 41 shows the catch in position to stop the movement of the stock at its point *A*. The stock is feeding to the right. The conical-pointed pin *B* is pushed by the spring *S* so that it engages a conical depression *C* in the end of the catch. By this means the toe of the



Figs. 40, 41 and 42. Positive Heel and Toe Latch which prevents the Stock from moving more than One Blank per Stroke of the Press

catch is pressed against the die. As the punch descends to cut the next blank, an adjustable screw on the punch plate presses on the top of the catch at *D* and causes the heel to lower and the pin *B* to disengage the notch *C*. The position of the latch is now shown by Fig. 42. Its heel *E* has been lowered into the hole left by the previous blank. It is held in this position by the pressure of the point of *B*. While this is sufficient to hold the catch in its new position, it offers

but little resistance to its return to its original position. The stock may now be moved along. The metal *K*, left between two successive blanks, engages the heel *E* of the latch and lifts it easily. This causes the notch C to engage with the pin *B* and the catch snaps back into its first position. The toe *A* falls into the new opening *R*, and *M* comes to a stop against it. Since the metal *K*, between two successive blanks, cannot pass the heel of the latch without raising it, and since the heel *E* cannot rise without lowering the toe *A* far enough to catch the stock, it is evident that the action is positive. Hence the stock cannot jump ahead faster than one blank at a time. In constructing a stop of this kind, care must be taken to allow under the heel *E*, Fig. 41, but little more height than the thickness of the stock. The length of the catch from toe to heel should be less than the opening left by one blank; then there will be no difficulty in starting the new ends of strips or coils. If necessary, however, the catch may be made so as to measure a little less than two or more openings in the

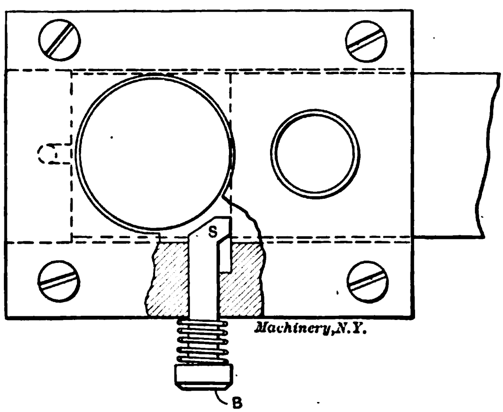


Fig. 43. Starting Device for a Gang Die

stock. In such a case the catch would have to be tripped by hand until the first piece of stock *K*, between two blanks, had passed under the heel *E*. This would cause delays which would amount to considerable in the case of strip stock.

This style of stop-pin has been used successfully with gang dies cutting blanks from brass 1/32 inch thick, and cold rolled steel 1/64 inch thick. In the case of the steel blanks, reels were used and the scrap was wound on a reel as it came from the die. By keeping the proper tension on the scrap, the stock was pulled through the die and kept against the stop-pin. Four thousand blanks per hour were made by this means. In view of the thin stock used and the fact that the dies were of the combination type, this was considered very good. The stop-pin had to be set accurately because the thin stock prevented the pilot-pins from shifting it much in aligning it. Other precautions taken on account of the thin stock were to make the toe broad and to fit the stripper close to the front edge of the toe.

The Gang Starting Device

The devices so far described serve to stop the stock when it has passed the blanking punch. But there are many cases where two or more operations are performed on a piece before it reaches the blanking die and the usual stop-pin. The operator usually gages the proper positions by watching the end of the stock through openings in the stripper, but it is better to have temporary stop-pins that can be used for that purpose. Fig. 43 shows a starting device for a gang die with two punches. When starting a strip the button *B* should be pressed. This brings into action the temporary stop *S*, which locates the stock properly for the first operation. It is then released and springs back out of the way. The stock is then advanced to the regular stop-pin. As many of these side stops may be used as are necessary. Not only do they save annoyance and time, but they add to the life of the dies by preventing the partial cuts due to the stock entering too far at the start.

CHAPTER VI

PRACTICAL EXAMPLES IN DIE DESIGN*

A few years ago, what is now the Providence Mfg. & Tool Co., of Providence, R. I., began the manufacture of a mechanical accountant, the invention of Mr. Turck, the present superintendent of the shop. Mr. Turck's experience, so far as shop work and tool design is concerned, had not been in the direction of die-making, so that in equipping the new plant for the manufacture of the accounting machine he was at first hampered by his lack of knowledge on this subject. The die work required was of a high order. The construction of machines of this type is often such that errors are cumulative. Several similar parts are used, attached to each other in series, for instance, in such a way that if the holes by which they are riveted to each other are slightly wrong in their dimensions, the error will be multiplied by the number of parts. The machine depends for its operation quite largely on the action of pawls upon fine ratchet teeth, and on the meshing of fine pitched gears and toothed segments with each other. The effect of cumulative errors in such circumstances would be to throw these fine pitched ratchets and gears out of step, and make the operation of the machine impossible. Long leverages are also a disturbing factor. When a long, slender member is located by two rivet holes close together, it takes careful work in punching those rivet holes to bring the parts into alignment. In the following some very interesting tools, used mainly for blanking purposes, but also for bending and other operations necessary to complete the product, are shown.

In the halftone in Fig. 44 are shown a number of press-made parts. Some of these are interesting in themselves, while others are remarkable principally for the methods used in producing them. Part No. 12, for instance, is a very simple piece, but the punch and die used in piercing the holes, while not unusual so far as surface appearances go, will serve well to illustrate some of the original practices of this shop. This punch and die, shown in Fig. 46, perform the simple operation of punching the nineteen small holes in the blank, which is located over die *A* by the carefully fitted aperture in jacket *B*. The punch is composed of a body *C*, a cast-iron holding plate *D* in which the small punches *E* are driven, a stripping plate *F*, held as shown, and forced outward by the compressed rectangular ring *G* of rubber behind it.

The Construction of a Piercing Punch with a Novel Stripper Plate

The making of this punch and die follows, in general, the order given below. Stripper *F* is first made of tool steel. The holes for the dowels *H* are next drilled. Then the holes through which punches *E* pass are laid out from a model or drawing, as the case may require, and drilled to a *larger* diameter than the punches which are to pass through

* MACHINERY, June, 1907.

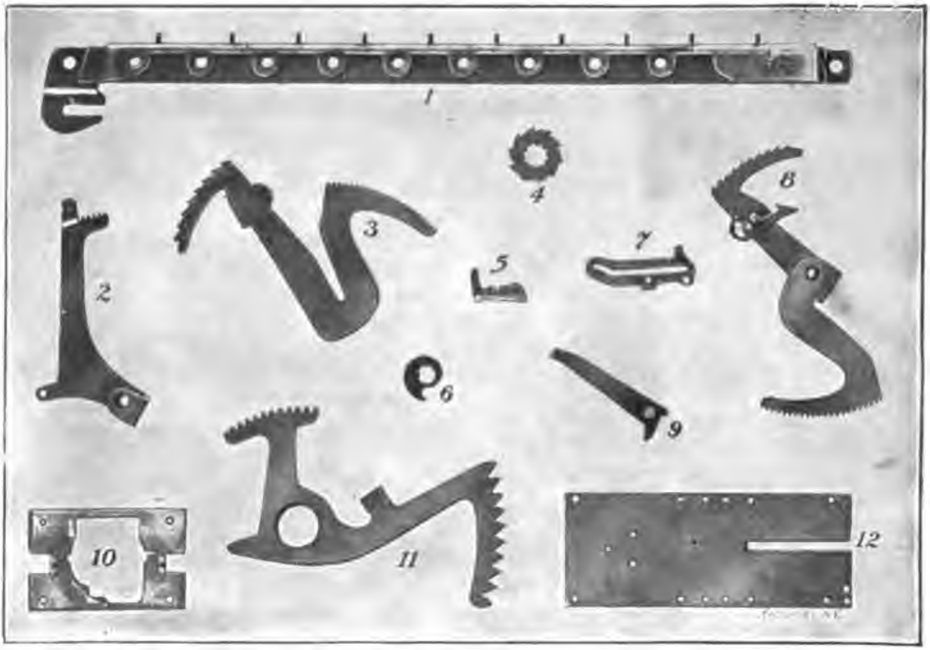


Fig. 44. Some Examples of Good Press-work

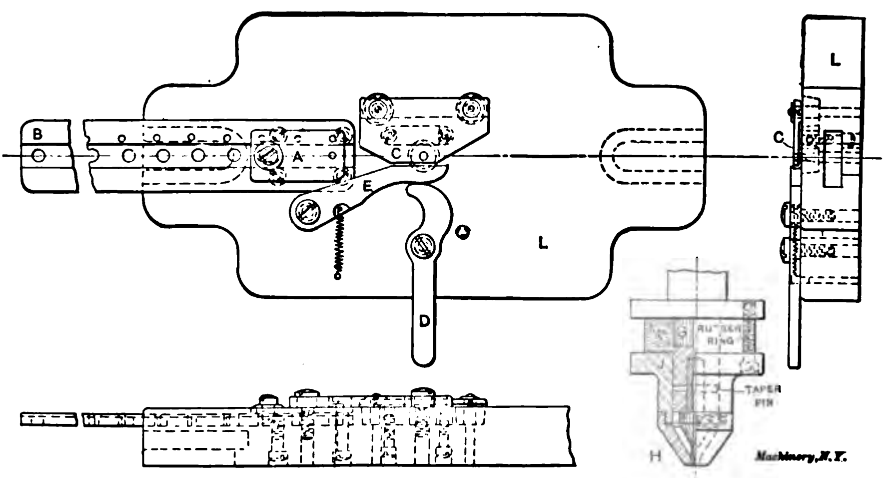


Fig. 45. Construction of Die for Double Punching

them. After these holes have been drilled, the plate is hardened and ground, and the holes for the punches are filled up again by driving into them plugs of tool steel wire, of suitable size. The location of these holes is now laid out again on plate *F*, and this time very carefully; then they are finished to the exact size, or slightly below, if they are to be lapped. Since the body of the plate is hard, it cannot cave in or wear as it would if left soft. A full bearing on the stock to be blanked is absolutely necessary if the work is to be well done. The plugs allow the plunger holes to be located after the hardening of plate *F*, thereby preventing displacement from the heat treatment. To the stripper plate are now riveted the four dowels *H*, which enter holes in the stripper rim or "collet" *J*, and locate the plate. Small round-headed set-screws bear on pins *H* and hold *F* and *J* together. Punch holder *D*, of cast iron, is machined to fit closely in collet *J*, and the holes for the punches are transferred to it from stripper plate *F*. The punches *E*, made of tool steel wire, are now driven into the holder, headed over at the back side, and ground flush. The punches may then be hardened in the usual manner. Before being assembled on the punch body *C* with the rubber spring *G*, a hardened steel backing *K* is inserted between *D* and *C* to take the thrust of the hardened punches.

The rubber spring *G* is cut from sheet stock and may be made either from separate strips built upon each of the four sides of the punch, or from rectangular rings, if that can be done without wasting the stock. Screws *L* are adjusted to bring the face of the stripper flush with the faces of the punches, after which headless set-screws *M* are screwed in to make the adjustment permanent. Screws *L* may then be taken out and replaced without losing the adjustment. The punch holder *D* and pad *K* are held to the holder by screws *N* and dowels *O*.

A Piercing Die with Inserted Tool Steel Plugs for Cutting Edges

The body *A* of the die is made of soft steel or cast iron. In this body are driven standard taper plugs of tool steel of suitable size, and so arranged as to be in position to furnish a tool steel material for all the actual cutting surfaces of the die. In the case shown in Fig. 46, nine of these plugs are used, carrying from one to three holes each. In making the recesses for these plugs standard tools are used. The seats are first drilled nearly to size, and then finished with a tapered end mill or counterbore, which is kept carefully ground to the proper dimensions, so that when the plug is driven in until it binds tightly on the taper, it will also seat on the bottom. These various plugs *P* are prevented from turning in the holes by dowel pins *Q*, in most cases, or, where the plugs run into each other (as shown in two cases in the die here described), by the interlocking of the flat abutting surfaces. These precautions make it possible to remove the plugs at any time and return them accurately to their original positions.

The die plate *A* having been fitted with its plugs as described, the holes in stripper plate *F* are now transferred to it by any suitable

means, all these holes being received in the tool steel plugs as explained. The plugs may now be removed, to be hardened and lapped separately. The clearance holes for the scrap are drilled, and the plugs are returned to their proper places. The jacket *B*, which locates the blank on the die, may, if desired, be punched from stock of suitable thickness by the blanking die used for making the blank to be

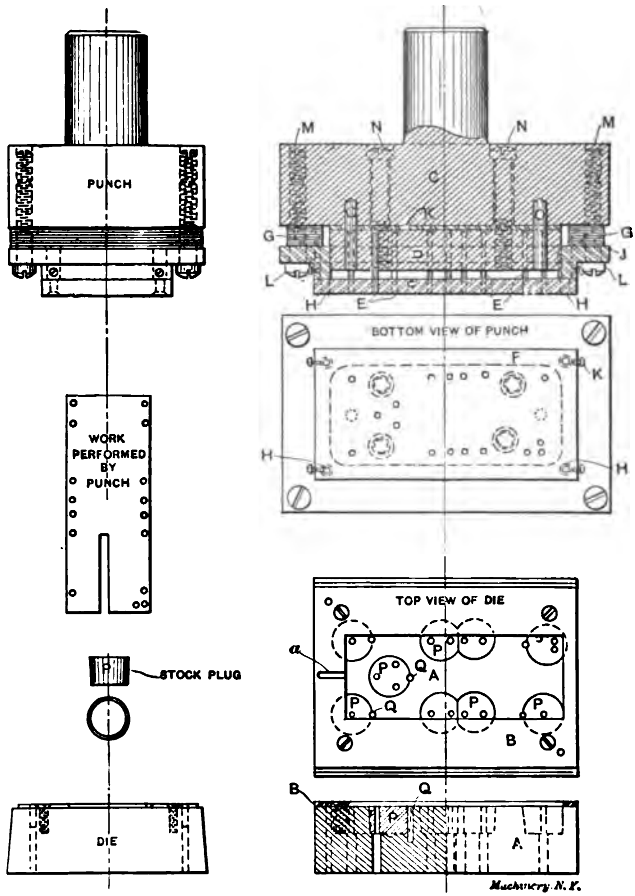


Fig. 46. A Piercing Punch and Die involving some Original Ideas

operated on in this piercing die. The edges of the opening are then merely filed enough to allow the work to enter and be withdrawn easily. A slanting groove, as shown at *a*, is cut with a round file into the jacket at one end to permit the insertion of a pick or awl to remove the work.

The points of interest in this die are: The rubber-backed stripper plate; the use of a soft stripper plate bushed in the manner described

with hardened tool steel; and the insertion of plugs of tool steel in a soft die block to form the cutting edges of the die.

The rubber spring has proven very satisfactory. It will last for a number of years in dies having ordinary use, if it is not exposed to oil and other deteriorating influences. Being in the upper member, there is little likelihood of its being spoiled in this way. The use of this stiffly spring-supported stripper plate gives a punch and die of the design shown all the advantages of a sub-press, so far as concerns the ability to punch small holes in thick material and leave thin walls of metal between open spaces in the punching. As evidence of the ability to do work of this kind with a punch and die of the style just described, parts 7 and 10 in Fig. 44 may be particularly noted. Here the holes are considerably smaller in diameter than the thickness of the stock, and the internal spaces have been punched so close to the edge, in places, that the remaining section is narrower than it is thick.

The method of bushing the stripper plate by drilling the holes large originally, plugging them with tool steel wire after hardening, and redrilling them to the proper size, makes it possible to harden the surfaces in contact with the work, without distortion of the dimensions between the holes. Plates of large size, even, are made in this way.

The advantage claimed for the method by which the stripper plate is made may also be claimed for the use of hardened plugs in a soft die body, since it is possible to harden these parts individually without changing their location with reference to each other. In addition, both of these schemes allow changes to be made in the dies with a minimum of trouble and expense. If it is desired to change the location of a hole in the die, the old plug may be removed and a new one inserted. In the same manner, new holes may be drilled in the stripper plate in which new tool steel wire plugs may be driven for new guiding holes for the punches, although the change is limited by the size of the plugs. This consideration is of considerable importance if the parts manufactured are subject to improvement from time to time. This provision reduces the expense of spoiled work as well, since it is not necessary to throw away an expensive press tool if one or two of the holes are wrongly located.

Rubber-backed vs. Sub-press Dies

It will be noted that part No. 12 in Fig. 44 (for which the punch and die just described were designed) is made in three operations. Under ordinary conditions, experience seems to indicate that this procedure is preferable to the use of the sub-press. The rubber spring-supported stripper plate, as just described, gives all the advantages of the sub-press, so far as ability to do fine work on thick stock is concerned. Slender punches are supported by the stripper in the same way as in the sub-press; the rubber spring holds the stripper so firmly onto the work that the distortion of thin stock is prevented. The sub-press certainly has the advantage of ease of setting in the machine, since it is not necessary to carefully line up the punch and die,

which are in permanent alignment. It is possible, however, that the high initial cost of the sub-press would in many cases more than pay for the extra wages of an experienced and careful man in setting up tools during the lifetime of the punch and die. It must also be admitted that work cannot be done as rapidly with the three sets of tools necessary for making the piece in the manner here described, as would be possible if a sub-press were used. The saving in first cost, however, and in the cost of subsequent operations, is believed to be sufficient in the case of the Providence Mfg. & Tool Co. to show a

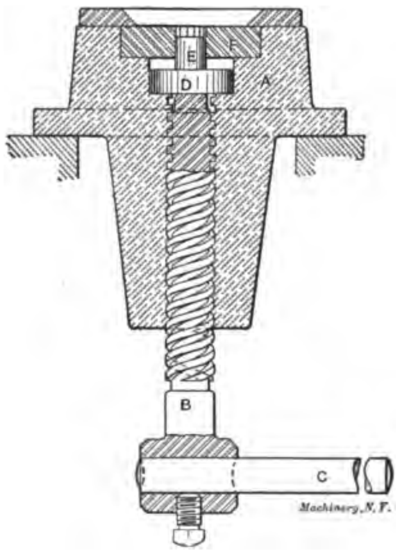


Fig. 47. Die for Bringing up Drawn-down Corners

balance on the right side of the sheet for the simpler form of press tool. It should be said in this connection that this firm freely makes and uses the sub-press die.

The Thickening of Corners Drawn out in Blanking

An operation of particularly great interest is a coining process used for reshaping the points of gears, ratchets, etc.—such parts, for instance, as are shown in samples 4 and 6, Fig. 44. In such a piece as No. 4, whatever the design of the die, the blank produced will be found to have the points drawn down thinner than the stock thickness. To bring the part back to uniform thickness with sharp points, the device shown in Fig. 47 is used. Here we have an attachment to a hand screw press. The body *A* is fastened to the bed of the press. The screw *B* projects through the bed and carries at its lower end a handle *C*, which is adjusted to one side or the other to bring it in position to be swung by the foot of the operator. In a counterbore in body *A* is seated the plug *D* and the ejector *E*. *D* and *E* are forced upward by the action of screw *B*. At *F* is a die, given the shape de-

sired for the outline of the finished part; it is slightly enlarged, however, for a short distance at its upper end. The part as it leaves the blanking press is purposely made a little large in outline at the points where the thinning occurs, due to the drawing out of the stock. When the piece is inserted by the operator in the upper end of this tapering die, the extra metal thus provided is forced inward to thicken the points to the required amount as the punch is brought down upon the work by the hand of the operator. When the piece has been forced to the bottom, it is clamped between the plane surfaces of ejector *E*

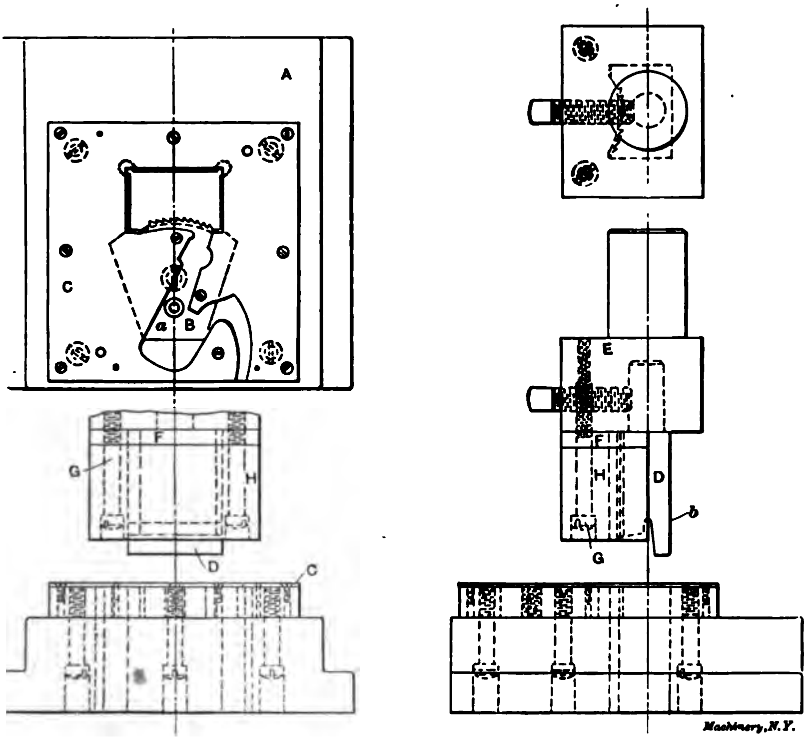


Fig. 48. Example of Type of Die used for Shaving

and the punch above it (not shown), and the metal is forced to flow to that part of the blank where it is most needed. The result is a flat ratchet with plane faces and uniform thickness. It will be understood, of course, that during this coining operation ejector *E* and plug *D* seat in the counterbore in body *A*, screw *B* being lowered out of contact. A push of the operator's foot on handle *C* brings the ejector up again until the piece is forced out of the die. The thread of the screw is of such a steep pitch that the screw will return again by its own weight.

The comparative slowness of operation resulting from the use of a hand and foot power press and hand feeding is, in a measure, char-

acteristic of this shop. It is the belief of the superintendent that better results can be obtained at times by methods like that shown, than by more "modern" ones. The aim is, through careful workmanship and careful inspection, to have the parts so nearly right when assembling time comes, that no fitting will need to be done in the assembled machines. No fitting is, in fact, allowed. Certainly the method described for striking up the corners of these ratchets is a much less dangerous one than would be the case if a power press were used, so the idea has its advantages, so far as safety is concerned, at least.

A Typical Shaving Die

In such parts as are shown at 3 and 11 in Fig. 44, the ratchet teeth and gear teeth are only roughed out in the blanking die, being finished by a second cut or "shaving" process. A typical die and punch for

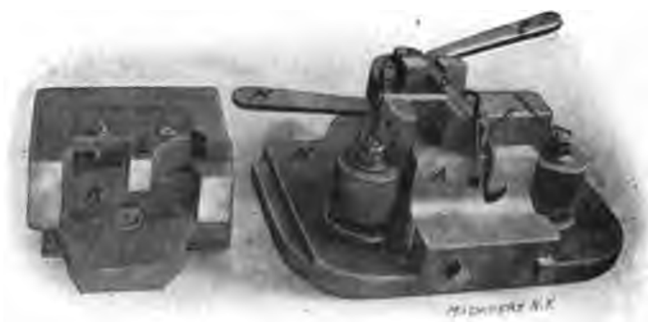


Fig. 49. Bending Attachment with Removable Die. Operation Completed

this operation are shown in Fig. 48. Here, as in Fig. 46, the work is held by a rubber spring backing while the punch is at work. The die is made of a soft body *A*, in which is inserted the hardened piece *B* carrying the cutting edges which are to form the ratchet teeth on the work. This piece *B* has its teeth cut on it in the milling machine, the hole at *a* serving to center the piece for this operation. This gives assurance that the teeth will be properly spaced, and cut accurately to the proper radius. A rectangular opening with carefully machined sides is made through the die block *A*. Into this opening the toothed cutting edges of piece *B* project. As in Fig. 46, a "jacket" *C* is provided for locating the work over the cutting die. The punch *D* is set into a holder *E*, which in turn is fastened in the ram of the machine. A projecting guiding surface, *b*, on the punch, enters the rectangular opening in the die and bears against it on the back and sides. This keeps the cutting surface of the punch up to its work against the cutting edge of the die. As shown, the cutting edge of the punch is beveled. This gives a slight top rake to the edge, and produces a shearing cut as well, the outer corners coming into action before the

center of the outline reaches the stock. The rubber spring backing at *F* is held by screw *G* between the pressure block *H* and the punch holder *E*. It performs the same functions as the stripper plate in the other die.

Bending Punchings to Provide Double Bearings

It will be noticed that samples 1, 5 and 8 in Fig. 44 have been made on the principle of bending the punchings to give a double bearing at pivotal points, the long bearing insuring lateral steadiness of the part without making it necessary to resort to the use of castings with long hubs. This principle is carried out throughout the calculating machine which is this firm's principal product. In some cases, espe-

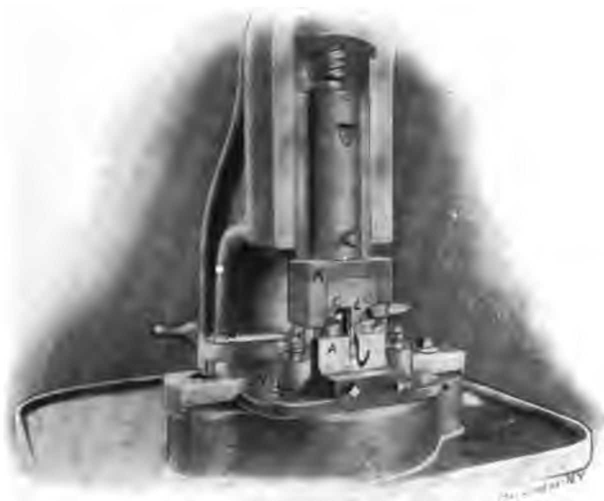


Fig. 50. Bending Device in Use in Screw Press

cially where the pivot holes are punched previous to bending, as is the case in sample 8, very accurate work must be done in the bending to bring the part to exactly the right form. In the sample referred to, for instance, the ratchet teeth on one side and the gear teeth on the other must bear a definite relation to each other, and to the axis about which the part rotates. The bending tools by which the forming operation is performed for this part are shown in the halftones in Figs. 49 and 50 and the engraving Fig. 51. Referring to Fig. 51, the blank for part 8 (shown at No. 3 in Fig. 44 before the piercing of the pivot holes) is laid on top of former *A*, where it is located by the pins *B B* which enter the pivot holes. In this position the part lies between the fixed jaw *C* and the movable jaw *D*, which are then clamped together on the blank by bringing handle *E* to the position shown, where its wedge-shaped cam surface *b* has entered between the long ends of the jaws *D* and *C*, and brought the outer ends together.

The jaws *D* and *C* and lever *E* are all attached to the holder *F*, which is a sliding fit on three vertical posts *G*, fast to the base *H* of the fixture. Slide *F* is held to the upper extreme of the travel against the lock nuts and washers at the top of posts *G* by spiral springs *J* at each

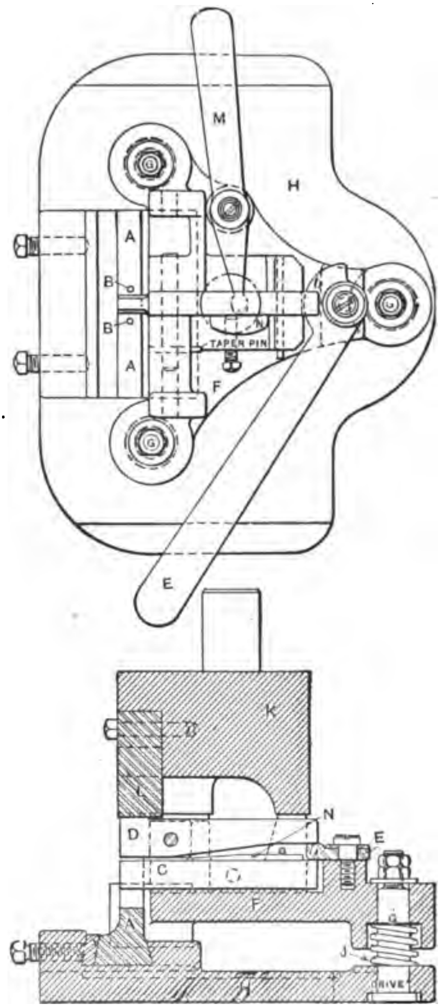


Fig. 51. Construction of Bending Attachment

post. These parts are shown to good advantage in the halftone, Fig. 49. *K* is a plunger mounted in the ram of the press. It bears on finished projections on slide *F* at three points as shown, while the hardened part *L* bears on the top of lever *D*, directly over the work. When *K* and *L* strike slide *F* and lever *D* in their descent, they carry with it

the slide and its attached levers, and the work as well, against the slight resistance of springs *J*. The work grasped between the levers is thus carried down through the opening in die *A*. This action serves to bend the part to the form desired. Fig. 50 shows the operation completed. As shown, this work is done in a hand screw press. This is another example of manufacturing methods which at first sight seem rather crude, but which have proved, in the opinion of the superintendent of this shop, to be most satisfactory, his contention of greater accuracy and more uniform results from such methods applying particularly in the case of forming operations of this kind.

The piece is ejected from the tool at the completion of the bending by lever *M*, which thrusts forward the ejector *N*. This ejector is at its working end slightly less in thickness than the stock of the punching operated on, and is thus able to enter freely between the jaws

Fig. 52. Double Punching Die shown in Fig. 45

and eject the work. In this tool, members *A*, *C* and *D* are changed for different parts, the rest of the structure being the same, and serving for a number of different operations.

A Die for Double Punching

In the case just described, where double bearings occur, the holes are punched before bending. This is not always the case, however. In samples 1 and 5 in Fig. 44, the parts are first bent and then punched, the operation being performed in a very interesting way. The punch descends and makes the hole in the upper thickness of the stock. Continuing through an intermediate die, and carrying before it the punched-out stock, it arrives at the second or lower thickness of stock. The continued movement of the punch then presses the little plug of punched-out metal through the lower thickness of stock, and this forms the second hole. Strange to say, it has been found in practice that this second hole is generally the better one of the two, even though it is made with a soft plug of steel instead of with a hardened punch.

The engraving Fig. 45 and the halftone Fig. 52 show the double punching tools used in making the pivot holes in sample 1, Fig. 44. This, it

will be seen, is a progressive operation, all the parts in the lot being punched for one of the holes, after which the die is altered and the next hole in order is punched in all parts—and so on. The piece to be operated on is located lengthwise by slipping it over a gage pin in sliding block *A*, which may be adjusted to any position on slide *B* to suit the hole it is desired to punch at the time. Being located on block *A* in the manner described, it is swung around until the intermediate die *C* enters the channel formed by the two sides of the work. Cam lever *D* is then swung to the position shown in the engraving, where it has brought clamp lever *E* against the stock, holding it firmly in position for the operation. The punch *F* is a simple turned piece of hardened steel, held by a taper pin in punch holder *G*. It is surrounded by a stripper *H* which is screwed to a holder *J*, backed by the usual rubber spring at *K*. This serves to hold the work firmly during the operation, and strip the work from the punch when it returns to its upward position. As before described, the punch in its descent breaks through the upper thickness of stock, carries the plug of soft metal thus formed before it until it comes in contact with the lower thickness, where it forces the plug through, and forms the lower hole. It will be noticed that intermediate die *C*, though held firmly, so far as displacement horizontally in any direction is concerned, is yet provided with a rocking face where it bears on the body of the die *L*. This arrangement takes the strain of the punching from the slender intermediate die, which is thus bent downward until it is firmly supported by the stock of the part being worked on beneath it. For removing the work after the operation, an ejector *M* is provided, with a handle *N*, which operates in a way which will be easily understood from an inspection of Fig. 52. It is not shown in Fig. 45, having been added at a date later than that of the drawing from which this cut was made.

Practice in Hardening Punches

Blanking punches are hardened in this shop in a way that is originated here and not practiced elsewhere, at least not to any great extent. After the blanking punch has been cut into the female portion of the die, and finished ready for hardening, it is placed in the fire and brought to a slightly lower heat than ordinarily used for hardening clear through. Cyanide is then deposited on the parts of the tool to be hardened—that is, on the periphery of the cutting edge. It is allowed to "soak in," it sometimes being necessary to apply cyanide two or three times, depending on the size and bulk of the punch. It is then again brought to the proper heat, which should be a little lower than is ordinarily used for hardening clear through. Then it is quenched in oil. With large and bulky pieces it is first necessary to immerse the work in water as a preliminary cooling operation. This immersion should merely be a dash into the water and out again, after which the piece is put into the oil until cooled.

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CHAPTER I

ELEMENTARY PRINCIPLES OF CONE PULLEYS AND BELTS*

Everyone knows that cone pulleys are usually made with regular steps; that is, if it is one inch from one step to the next, it is also one inch from the second to the third, etc., the reason being that when the centers of the shafts on which the cones run are a fair disance apart, the belt will pass very nearly half way around that part of each cone on which it is running, and the length of the belt will consequently be approximately equal to twice the distance between the shafts, added to half the circumference of the grade of one of the cones on which it is running, and half the circumference of the grade of the other cone on which it is running. As the steps are even, the half circumference of any two grades of each cone will, when added together, produce the same result. For example, if we had two cones, the diameters of the several grades of which were 6, 8, 10 and 12 inches, it is evident that the sum of half the diameters taken anywhere along the cones, as they would be set up for work, would in every case be the same. If the diameters are the same, it follows that the circumference must also be the same, and, of course, that half the circumference must be the same, so that when the centers of the shafts are a fair distance apart, and the difference between the largest and smallest step of the cone not too great, the same belt will run equally well anywhere on the cone, because it runs so near half way around each grade of the two cones on which it is running, that the slight difference is within the practical limit of the stretch of the belt.

But when the shafts are near together, and when the difference between the largest and smallest step of the cone is considerable, the belt is not elastic enough to make up this difference. Fig. 1 shows a three-step cone, the grades being 4, 18, and 32 inches diameter, respectively, there being a difference of 14 inches on the diameter for each successive grade, and the step being therefore 7 inches in each case. Of course, it is not likely that such a cone as this would be made for practical use, but it is well to go to extremes when looking for a principle. Now, it is evident that two cones, even if like the one shown in the cut, were set up far enough apart, they would still allow the belt to run very nearly half way around each grade of the two cones, the angularity of the belt would be slight, and the length of belt would therefore still be as mentioned above.

But (again taking an extreme case) by reference to Fig. 2, which is intended to represent a belt running from the largest grade of one cone to the smallest grade of the other cone, we see that the belt runs three quarters of the way around the large pulley, and only one quarter

* MACHINERY, April, 1895.

of the way around the small one, the distance between the shafts in this case being 19¾ inches.

The length of this belt will evidently be equal to three quarters of the distance around the large pulley, plus one quarter the distance around the small pulley, plus the distances *A* and *B*, which we find to be each 14 inches. The circumference of a 32-inch diameter pulley is 100½ inches, and the circumference of a 4-inch diameter pulley is 12½ inches (near enough for our present purpose); three quarters of 100½ is 75⅜, and one quarter of 12½ is 3⅛; the length of a belt, then, to go around a 4-inch pulley and a 32-inch pulley, running at a distance of 19¾ inches apart, is 75⅜ plus 3⅛ plus 14 plus 14; total, 106½ inches.

Now, let us take the middle cone, when the belt is running on two pulleys, both 18 inches diameter (see Fig. 3), and, of course, the same

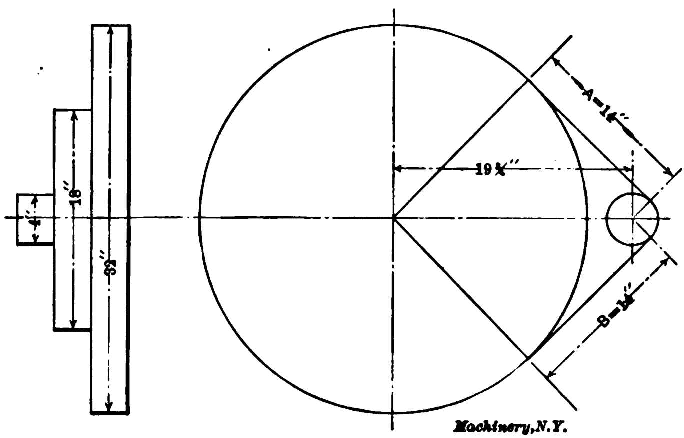


Fig. 1 Fig. 2

distance apart as before. The circumference of an 18-inch pulley is 56½ inches, and half the circumference of two 18-inch pulleys is evidently the same as the whole circumference of one 18-inch pulley; the length of belt in this case will then evidently be 56½ plus 19¾ plus 19¾; total, 96 inches. It is therefore evident that a belt long enough to run on a 4- and 32-inch pulley, 19¾ inches apart, is 10½ inches too long to run on two 18-inch pulleys 19¾ inches apart, and, of course, it is therefore 10½ inches too long to run on the middle grades of such a cone as we have under consideration.

The thing to do, then, is to make the middle grades of these cones (or the two 18-inch pulleys) enough larger than 18 inches diameter to just take up this 10½ inches of belt, and if this were the only case we had to deal with, it would be very easy to settle it by saying that as half the circumference of two 18-inch pulleys is the same as the whole circumference of one 18-inch pulley, we should make the two 18-inch pulleys enough larger in diameter to make an additional circumference of 10½ inches; and as 3⅜ inches is nearly the diameter of

a $10\frac{1}{2}$-inch circumference pulley, by making the middle of both cones 18 plus $3\frac{3}{8}$ inches diameter (that is, $21\frac{3}{8}$ inches diameter) our trouble would be ended in this particular case. It is easy enough to see, by looking at Fig. 2, that the belt being obliged to go three quarters of the way around the large pulley, is what makes it so much too long to go around the two middle pulleys, where, of course, it goes but half way around each. But, of course, what we want is some way of calculating the diameters to turn any pair of cones, running at any distance apart.

If we were to draw these same 32- and 4-inch pulleys twice $19\frac{3}{4}$ inches apart, and then three times $19\frac{3}{4}$ inches apart, and so on, until we got them far enough apart so that the belt would practically run half way around each, and should calculate the diameter of the middle grade of the cone to fit each distance, we would probably formulate a rule that would work for any distance apart, with this particular cone; but as it is evident that the further apart the cones are to run, the nearer to the nominal diameter of 18 inches must the middle of

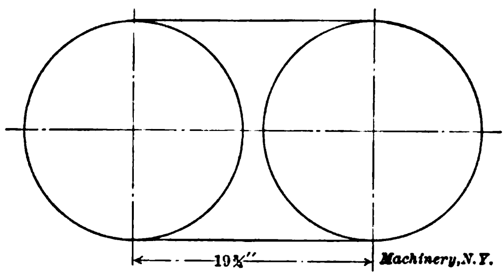


Fig. 3

the cones be turned, so also must it be evident that the less difference between the largest and smallest diameter of the cone, the less must also be the excess over nominal diameter of the middle of the cones.

Any method, then, of calculating such problems must take both of these things into consideration. The nominal diameter of the middle of any cone will be equal to half the sum of the diameters of the largest and smallest part respectively. This is almost self-evident, and no proof of it is necessary in this connection. What we want, then, is some way to find out how much larger than the nominal diameter to turn any one cone or cones to fit the conditions under which they are to run. The following formula is the result of a thorough investigation of this subject by Prof. Rankine, and has proved itself to be practically correct in the shop, as well as satisfactory to those mathematicians who are competent to criticise it. This formula is:

$$R = \frac{R_1 + R_2}{2} + \frac{(R_1 - R_2)^2}{2\pi C}.$$

This formula translated into plain English means that the radius of the center of a cone will be equal to the radius of the smallest part, added to the radius of the largest part, and this sum divided by

2, and added to this the difference in radii between the largest and smallest part squared, and then divided by twice the center distance between the cones multiplied by 3.1416. That is, the first half of the formula gives the radius at the center of a cone, when the largest and smallest radii are known, and, of course, if the middle radius is equal to the smallest radius added to the largest radius and the sum divided by 2, it follows that the middle diameter is equal to half the sum of the diameters of the largest and smallest part, respectively, as mentioned before. The second part of the formula allows us to calculate how much larger than this nominal diameter to make the middle of a cone, no matter what the size or center distance.

Applying this formula to the case of the cones shown in Figs. 1, 2 and 3, we find the radius of the middle of the cone to be 10 6/10 inches, or, what is the same thing, the diameter to be 21 2/10 inches, which, in view of the extreme case under consideration, is very near the first result obtained (21⅜), and shows that the formula is perfectly safe in any case likely to occur in practice.

When this formula is reduced so as to express the numerical value of diameters instead of radii, it takes the following form:

$$\text{Diameter at center of cone} = \frac{D+d}{2} + \frac{(D-d)^2}{12\frac{1}{2}C},$$

the 12½ being the nearest value in plain and easy figures to which the quantity containing π in the original formula can be reduced.

Applying this simplified formula to the cone which we have been considering, it will be found that the middle diameter is 21 2/10, the same as by the original Rankine formula.

If a cone has five steps instead of three, it will be practically correct to add half as much to the nominal diameters of the second and fourth grades as was added to the middle grade, or, if it has four grades, add two-thirds of what is found by the calculation to the second and third grades (as there is evidently no middle grade). If more than four or five grades, add to each grade according to the same principle.

We have so far been considering two similar cones, but it often happens that one cone is larger than the other. In such case the problem becomes a little longer to work, and the length of belt necessary to go around each pair of steps of the cones must be used to find the diameters; that is, starting with one end of the cone, find the length of belt, and then calculate how much larger or smaller (as the case may be) than the nominal diameter it is necessary to make each grade, in order to make the same length of belt run properly.

Prof. Rankine has worked out a formula for the length of belt also, which, reduced to diameters, is as follows:

$$\text{Length of belt} = 2C + \frac{11D + 11d}{7} + \frac{(D-d)^2}{4C}.$$

That is, the length of a belt to pass around any two pulleys (and, of course, a cone is simply a set of pulleys) is the sum of the following quantities: First, twice the center distance of the shafts; second, 11 times the diameter of the larger pulley, plus 11 times the diameter

of the smaller pulley, and this sum divided by 7. This gives the nominal length of belt, or what would be practically correct if the center distance was fairly great; for the excess, the last part of the formula must be used, which is the difference between the diameters of the larger and smaller pulleys squared, and this result divided by 4 times the center distance.

Having found the length of belt to run on one end of the cones, and keeping this for a starter, we can easily find how much to add to, or take from, the nominal diameter of any other part of the cone to make the same belt run, as explained before. If, for instance, we find that the nominal diameters of the next grades that we try to bring the length of belt one-half inch shorter than the first calculation, we add enough to one or both diameters to make up one-half inch of circumference, which would be about 5/32 of diameter, and this could all be added to one pulley, or half of it could be added to each pulley, as convenient, and this would be practically correct.

CHAPTER II

CONE PULLEY RADII*

In the present chapter a method presented by Dr. L. Burmester in his "Lehrbuch der Kinematik," for the solution of the cone pulley problem, has been extensively treated. Dr. Burmester's method is entirely graphical, and is exceedingly simple in application. While it is not theoretically exact, it is, as will be shown later, much more accurate than practice requires.

In order to bring out more clearly the points which will come up in the case of open belts, let us first consider the simple case of crossed belts. It is a well-known fact that in this case the only calculation necessary in order to find the radii of the various steps is to make the sum of the radii of any two corresponding steps a constant. This may be shown in the following manner:

a = radius of step, driving cone.

A = length of belt from contact on driving cone to contact on driven cone.

b = radius of step, driven cone.

E = distance between centers of cones.

K = the constant sum of the radii of two corresponding steps.

θ = angle shown, Figs. 4 and 5.

Then

$$\sin \theta_1 = \frac{a_1 + b_1}{E} = \frac{K}{E}$$

$$\sin \theta_2 = \frac{a_2 + b_2}{E} = \frac{K}{E}$$

* MACHINERY, September, 1905.

Therefore $\theta_1 = \theta_2 = \theta =$ a constant.

Therefore the arc of contact on each pulley $= 180° + 2\theta =$ a constant.

$$\text{Also } \cot\theta = \frac{A_1}{a_1 + b_1} = \frac{A_1}{K} = \frac{A_2}{a_2 + b_2} = \frac{A_2}{K}$$

$$A_1 = A_2 = K \cot\theta = \text{a constant.}$$

$$\text{But length of belt} = 2A + \frac{180° + 2\theta}{360°} \times (2\pi a_1 + 2\pi b_1)$$

$$= 2A + \frac{180° + 2\theta}{360°} \times 2\pi \times (a_1 + b_1)$$

in which, as has been shown above, all the terms are constants, therefore length of belt is constant.

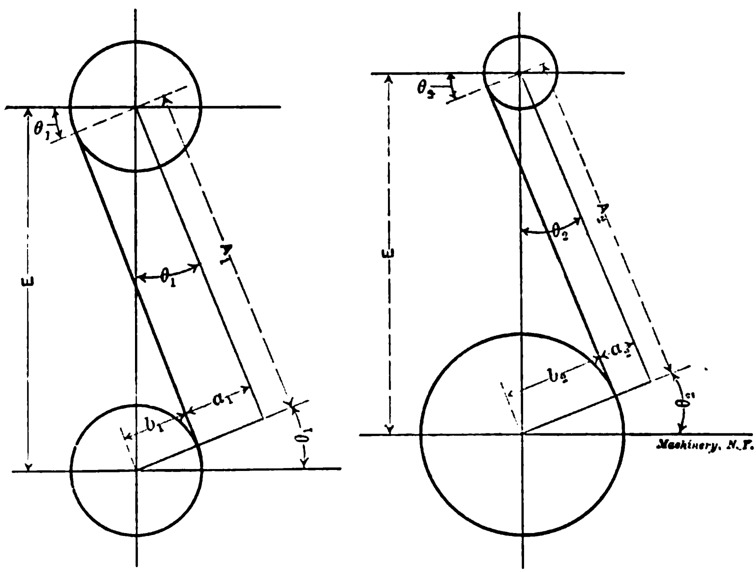


Fig. 4 Fig. 5

Note: The subscript applied to the letters denotes that the letters are used for the corresponding quantities in a special case; thus a_1, in Fig. 4, refers to a.

The radii of the various steps may be determined graphically by the following diagram (Fig. 6):

Draw a horizontal line from A, and also draw AC making an angle of 45 degrees with it. On this line lay off AS equal to the distance between the cone centers, using any scale most convenient, bearing in mind, however, that the scale adopted now must be used consistently throughout the diagram. At S erect the perpendicular TST' to the line ASC. From some convenient point on AC, as D, drop a vertical equal to some known radius of the cone a, as DE, and then

from E measure back on this vertical the radius of the corresponding step on cone b, as EF, and from these points E and F draw lines parallel to ASC. From the point G, where the line FG intersects the line TST' drop a vertical. This will intersect the line EH in H. Through H draw the horizontal MN, O being the point where this line intersects the line TST'. Then, distances on the line MO may be taken to represent radii on cone a; and to find the corresponding radii on cone b erect perpendiculars at the extremities of these radii, producing them until they intersect the line TST'. These perpendiculars then represent the desired radii. It may be shown as follows that the sum of the two corresponding radii, as obtained from this diagram, is always a constant, and the diagram therefore satisfies the conditions for crossed belts.

Let MJ represent any radius on cone a, then JI represents the corresponding radius on cone b.

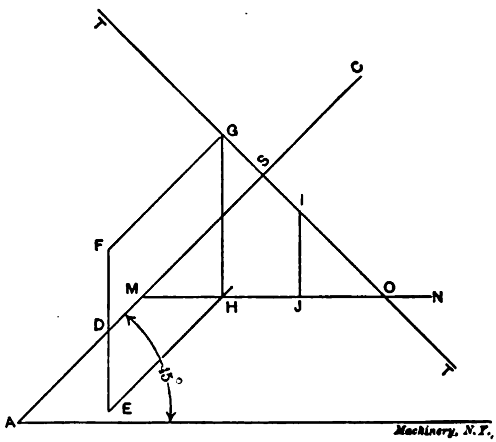


Fig. 6

The $\angle JIO = \angle JOI = 45$ degrees.

Therefore $JI = JO$.

Therefore $MJ + JI = MJ + JO = MO =$ a constant.

Dr. Burmester's diagram for open belts is a modification of the diagram just shown, the only difference being that the line TST' is replaced by a curve. This curve was determined by plotting a series of points, and after several pages of exceedingly intricate mathematics he arrives at the astonishing result that this curve can be replaced by a simple circular arc without any appreciable error.

The diagram is shown in Fig. 7, and may be drawn as follows: Proceed as in Fig. 6 until the line TST' is drawn, then lay off distance SK equal to $\frac{1}{2}$ AS. Next, with the center at A, and a radius equal to AK, describe the arc XY, and the diagram is ready for use.

In order to give an idea of the extreme accuracy of the diagram, let us observe the values obtained by Dr. Burmester in his calculations.

Let $R = AK$ (Fig. 7).

When $\theta = 0$, $R = E \times 1.11815172$.

$\theta = 15°$, $R = E \times 1.11806842$.

$\theta = 30°$, $R = E \times 1.11798671$.

$\theta = 45°$, $R = E \times 1.11803397$.

The value used for R in the diagram is $E \times 1.11803397$; so the maximum error of R occurs when $\theta = 0$, and is equal to $E \times 0.00011775$,

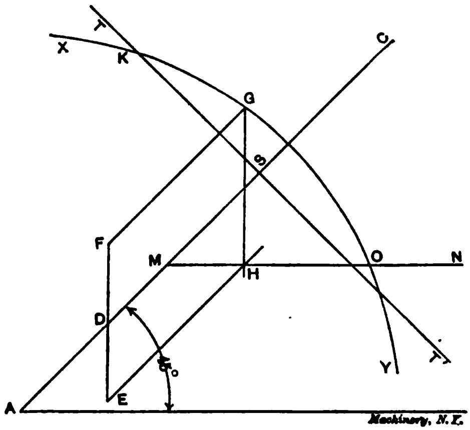


Fig. 7

which is much more accurate than the work of the most careful draftsman. Dr. Burmester gives values of R up to $\theta = 90°$, but as it is evident from Fig. 8 that it would be practically impossible to have a value of θ greater than 45°, the writer has omitted the other values.

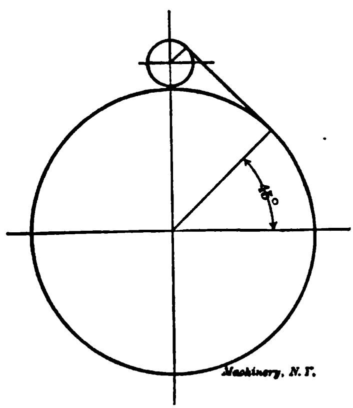


Fig. 8

In order to make the use of the diagram perfectly clear, let us solve the following problems:

Problem 1. Fig. 9

Given:

Distance between centers of cones $= 3' \ 4''$.

Diameters of driving cone, 4", 8", 14", 20".

Diameters of driven cone, X, X, 14", X.

Required:

All diameters of driven cone.

Lay out the diagram and determine the point *M* as previously directed. Now the radii of driving cone may be laid off as abscissas or ordinates, whichever happens to be the more convenient, as the results

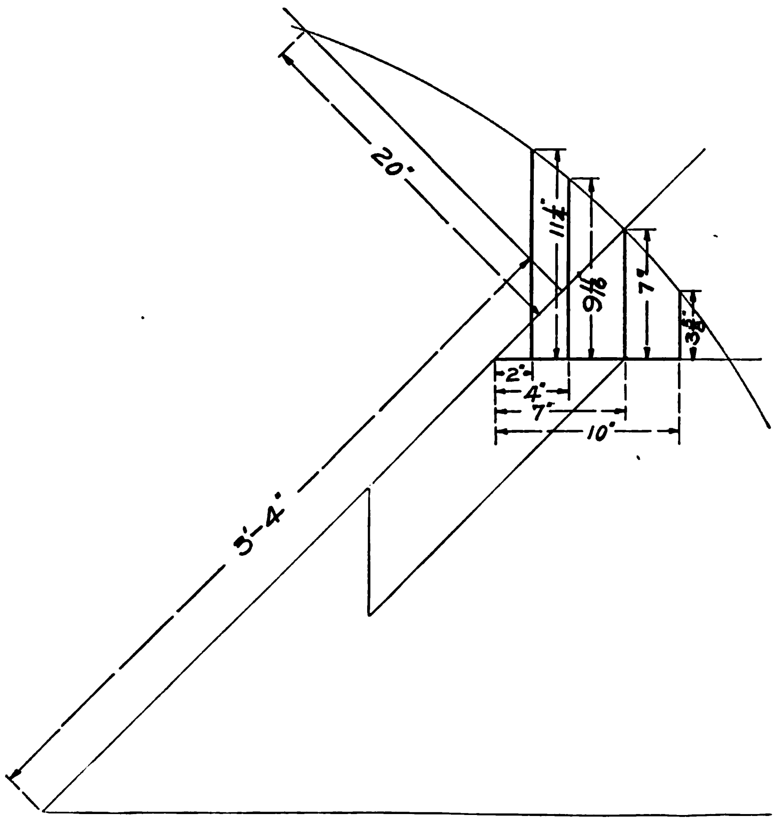


Fig. 9. Solution of Cone Pulley Problem when Diameters of One Pulley and Center Distance are Known

obtained will be exactly the same in either case. In this particular problem it is evidently more convenient to lay them off as abscissas. Then the ordinates erected at the ends of these abscissas will represent the corresponding radii of the driven cone. The problem is solved in Fig. 9 and the following results obtained:

Results:

Diameters of driven cone, 22½", 19⅝", 14", and 7¼".

This problem does not bring out all of the fine points of the diagram, so let us solve a more complicated one, in which the different steps of

the cone are to transmit given velocities.

Problem 2. Fig. 10

Given:

Distance between centers of cones = 3′ 4″.

Maximum velocity of belt (assumed) 30 feet per second.

R. P. M. of driving cone = 240.

Required:

Driven cone to make 100, 240, 400 and 580 R. P. M..

The maximum belt speed will be attained when the belt is on the largest step of the driving cone.

Therefore

$$\frac{2\pi a_1 \times 240}{12 \times 60} = 30; \quad 2a_1 = 28\tfrac{5}{8}''$$

$$\text{But } \frac{2b_1}{28\tfrac{5}{8}} = \frac{240}{580}; \quad 2b_1 = 11\tfrac{7}{8}''.$$

Now having obtained a value for a_1 and b_1, the point M on the diagram may be found. Next draw a line from M as MO, inclined so that any horizontal projection, as MN, will be to the corresponding vertical projection, NO, as the R. P. M. of the driver are to the R. P. M. of the driven; thus,

$$\frac{MN}{NO} = \frac{\text{R. P. M. of driving cone}}{\text{R. P. M. of driven cone}}.$$

Also from similar triangles

$$\frac{MN}{NO} = \frac{MN'}{N'O'}.$$

But we know that

$$\frac{\text{R. P. M. of driving cone}}{\text{R. P. M. of driven cone}} = \frac{\text{rad. of driven cone}}{\text{rad. of driving cone}}$$

Therefore MN' equals radius of driven cone, while $N'O'$ equals radius of driving cone, thus making, for this case, radii of driving cone vertical and of driven cone horizontal. The problem is solved in Fig. 10 and the following results obtained:

Results:

Dia. or driving cone, $28\tfrac{5}{8}''$, $25\tfrac{5}{8}''$, $20\tfrac{3}{4}''$, $11\tfrac{7}{8}''$.

Dia. of driven cone, $11\tfrac{7}{8}''$, $15\tfrac{3}{8}''$, $20\tfrac{3}{4}''$, $28\tfrac{5}{8}''$.

We have seen that the Burmester diagram is under all conditions much more exact than is required in practice; and a more compact, simpler, or quicker method of finding cone pulley radii could not be desired. An experienced draftsman should be able to solve a problem like No. 2 above in less than 10 minutes, while to obtain the same results by an analytical method would require as many hours. Results of sufficient accuracy can usually be obtained by making the diagram to half scale, although there is no reason for reducing the scale, unless the distance between centers is inconveniently large, and in that case

the results do not need to be so accurate, as the belt will stand more stretching.

Another graphical method for laying out a pair of cone pulleys is as follows: First draw straight line *A A*, Fig. 11, supposed to connect the centers of the cones to be laid out; then set off the centers of the cones *B* and *C* on line *A A* (full size is best); then bisect the distance

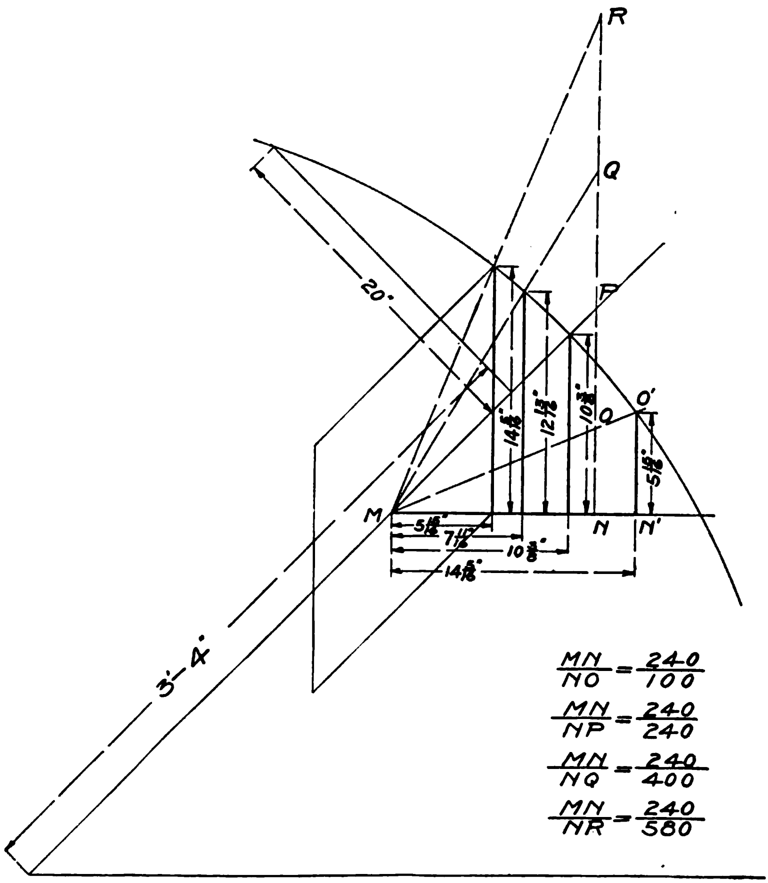


Fig. 10. Solution of Cone Pulley Problem when Velocity Ratios, Maximum Belt Speed, Center Distance, and R. P. M. of Driver are Known

between the centers of the cones and draw perpendicular line *D E*. Now assume the size of the two cones—say the largest is 25 inches and the smallest 3 inches diameter. Then draw a line tangent to the circles, or the line representing the inside of the belt *G*, which will intersect the line *D E* at *E*, and taking the point *E* for a center scribe the circle *F*. Then divide the circle *F*. commencing at the line of the belt *G*, into as many parts as needed, of a length to suit the required speeds.

Draw the other radiating belt lines through the point *E* and the divisions on the circle *F*, extending them toward the cone *B*, and they will be the inside of the other belt lines. Draw circles tangent to these

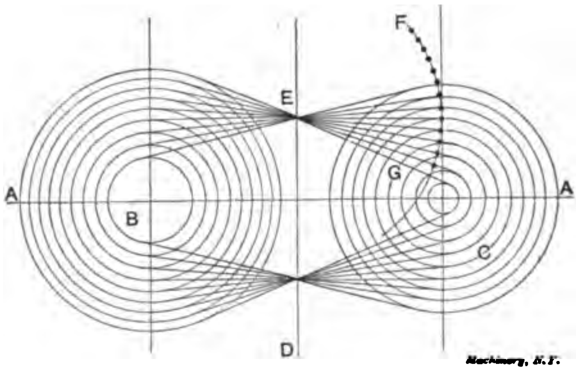


Fig. 11. Simple Graphical Solution of Cone Pulley Problem

lines. We now have all the diameters of the rest of the steps of the cone to match the first, and the belts will correctly fit all the steps. This is, of course, only an approximation rule. This method was contributed to the June, 1905, issue of MACHINERY by John Swanberg.

CHAPTER III

STRENGTH OF COUNTERSHAFTS*

There is scarcely a shop in existence which has not had a more or less serious accident from a countershaft some time in its history. It may have been caused by a heavy pulley running very much out of balance, or the shaft may have been bent in the beginning. Possibly the shaft was too light, or too long between hangers. The latter is responsible for most of the trouble, and is the one with which this discussion is principally concerned.

There are two methods in vogue for turning cones and pulleys; one is to set the rough casting to run true on the inside, and the other on the outside. This latter method makes a cheaper and an easier job, but when turned, it requires an enormous amount of metal to balance it. And here is the source of considerable trouble. We may balance a large cone perfectly on straight edges, but that is a standing balance only; and when the cone is put in place and speeded up to several hundred revolutions per minute, it shakes, and shows that it is decidedly out of balance. The trouble is that we have not placed the balance weights directly opposite, or in the plane of the heavy portion of the cone. The result is that neither weight, when rotating, has its counterbalance pulling in the same line, and, of course, the pulley is sure to be out of balance. All cones and all other pulleys which have a wide face should be set to run true on the inside before turning.

A certain countershaft failed because it had been welded near the center. The weld twisted and bent open, and some one was badly injured by the fall. A weld in machine steel is so very uncertain that it should never be trusted for such a purpose. The extra expense of a new shaft would not warrant the hazard of such a risk.

In the calculations which follow, the spring of the shaft is limited to 0.06 of an inch. There are plenty of countershafts which have been running for years with about this much spring. Now, from the general formula for the deflection of a simple beam, we have:

$$\text{The deflection, or spring} = \frac{W L^3}{48 E I}$$

in which W = the load at the center in pounds.

L = the length between center of hangers in inches.

E = the coefficient of elasticity = 29,000,000.

I = the moment of inertia of the cross-section of the shaft.

For a round shaft,

$$I = \frac{\pi d^4}{64} \qquad (1)$$

* MACHINERY, April, 1903.

in which d = the diameter of the shaft in inches. We then have:

$$\frac{W L^3}{48 E I} = 0.06 \qquad (2)$$

From (1) and (2), we have $\frac{64 W L^3}{48 E \pi d^4} = 0.06$, and

$$L = \sqrt[3]{\frac{0.06 \times 48 E \pi d^4}{64 W}} = \sqrt[3]{\frac{0.06 \times 48 \times 29{,}000{,}000 \times \pi}{64} \times \frac{d^4}{W}}$$

$$= \sqrt[3]{4{,}100{,}000 \frac{d^4}{W}} \qquad (3)$$

Fig. 12 shows a countershaft which is in actual service, and which is known to be all right. *A* and *B* are keyed to the shaft. *C* and *D* are

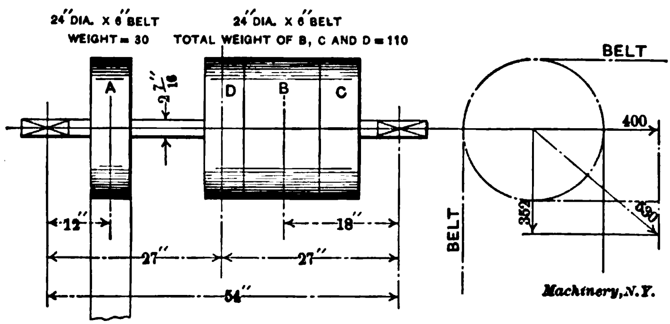


Fig. 12

loose pulleys arranged for open and cross belts.

A weighs 30 pounds, and *B*, *C* and *D* weigh 110 pounds. The belts run as shown in the figure. If *A* weighs 30 pounds, and the centers of the hangers are 54 inches apart, then by taking the left-hand hanger as the center of moment, we have $30 \times 12 = x \times 27$, when x is the weight at the center. Solving we find

$$x = \frac{30 \times 12}{27} = 13$$

In the same way, by taking the right-hand hanger for the center moment, we find that

$$x_2 = \frac{110 \times 18}{27} = 73$$

As to the belt pull, it is possible for a single belt to run up to 70 pounds per inch of width of belt, and a double belt can be taken at 100 pounds. As a double belt is used in this case, and as the slack side of the belt is very loose when the tight side is pulling its maximum, we will take the pull at the pulley $A = 6 \times 100 = 600$ pounds, and getting this in terms of a load at the center, we have

$$x_3 = \frac{600 \times 12}{27} = 266$$

and the downward pull is $13 + 73 + 266 = 352$.

The pull at the pulley *B* will be $6 \times 100 = 600$, and by transferring this to the center we have

$$\frac{600 \times 18}{27} = 400$$

The resultant of these two forces will be the diagonal of the force diagram, and is equal to 530 pounds, which is equal to *W* in the formula. Introducing these terms in equation (3) we have

$$L = \sqrt[3]{4{,}100{,}000 \frac{(2.44)^4}{530}}$$

and by solving we find $L = 65$, which means that for this condition of loading the countershaft would be safe even with the hangers 65 inches apart.

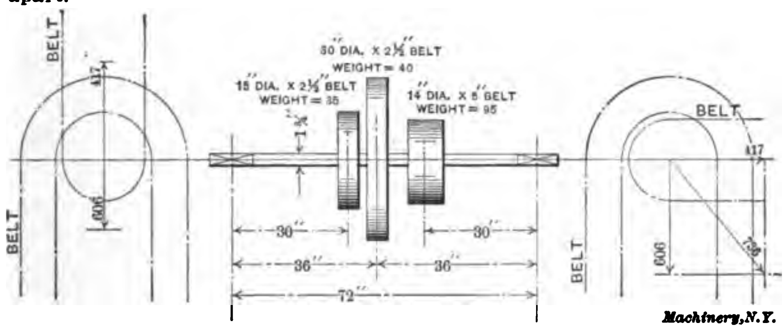


Fig. 13

Fig. 13 represents another countershaft taken from actual service. It is belted as shown on the left-hand view, and is running all right, although it looks rather flimsy, and one would consider it unsafe. Taking the moments of the weights of the pulleys and belt pull about the right- and left-hand supports, and finding the equivalent pull at the center, we obtain:

Weight at center due to pulleys =	148
Pull on 30-inch pulley = $2\frac{1}{2} \times 100$ =	250
Pull on 15-inch pulley = $2\frac{1}{2} \times 100$; $\frac{250 \times 30}{36}$ =	208
Total downward pull =	606
Pull on 14-inch pulley = 5×100; $\frac{500 \times 30}{36}$ =	417
Resultant downward pull =	189

Introducing this value of *W* in equation (3) we have,

$$L = \sqrt[3]{4{,}100{,}000 \frac{(1.75)^4}{189}} = 59$$

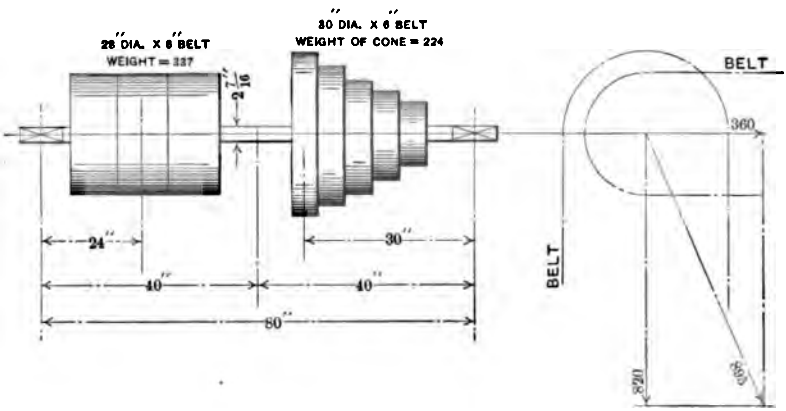
This is considerably less than the distance between the hangers, and it shows that it is not safe to place the hangers in this way. If the

belts ran as shown at the right-hand side of Fig. 13, we would then have:

Weight due to pulleys (as before) =	148
Pull on 30-inch pulley =	250
Pull on 15-inch pulley =	208
Total downward pull =	606
Horizontal pull on 14-inch pulley =	417

From these two forces we find a resultant of $W = 736$. Substituting this in (3) and solving as before, we find $L = 38$, which is the greatest safe distance between hangers for this condition of loading.

There are cases where one must have an extra long shaft in order to work in the pulleys, cones, etc., as shown in Fig. 14. Here the downward loads amount to 820 pounds, and the pull at right angles amounts to 360 pounds. The resultant 895 pounds $= W$.



Machinery, N.Y.

Fig. 14

Introducing in the formula we have

$$L = \sqrt[3]{4{,}100{,}000 \frac{(2.44)^4}{895}} = 55$$

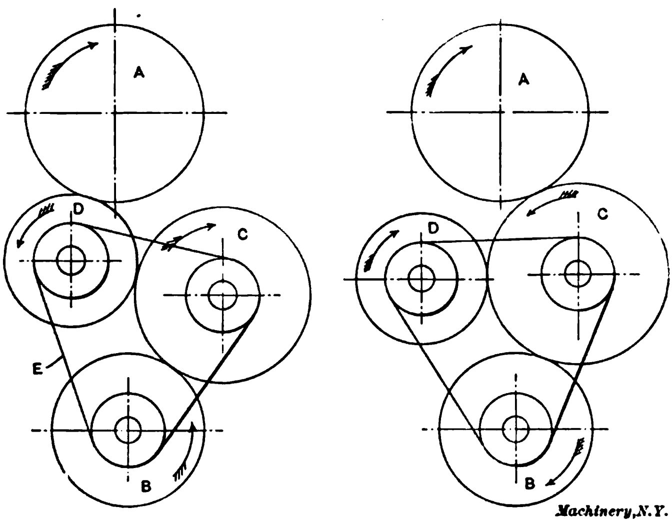
This means that for this condition of loading, the center distance should not exceed 55 inches, and since in this case it could not be made as small as this, the pulleys should be arranged for a third hanger.

In every case, therefore, where the centers are so far apart as formula (3) would indicate to be unsafe, a third hanger should be used. If all the flimsy countershafts had a third hanger added to them there is no doubt but that the number of accidents would be greatly diminished. In the above calculation the weight of the countershaft has not been considered, as it is usually very small. If the belts run at any other angle than that shown, the construction is made in exactly the same way, using the required angle instead of a right angle, the resultant of the two forces being used as W in the formula.

CHAPTER IV

TUMBLER GEAR DESIGN*

Of the different mechanisms that have been used in the machine tools of the past, one—the tumbler gear—could be found in some form or other in almost every machine. Its office, in most cases, was to reverse the direction of the feed. Fig. 15 shows the usual form in which it is found when used for this purpose. The gears *A* and *B* are to be connected so that motion may be transmitted from one, which runs constantly in one direction, to the other, which it is desired to



Figs. 15 and 16. Examples of Tumbler Gears

run in either direction. Suppose that *A* is the driver and runs as shown by the arrow. As connected, *A* drives *B* through the intermediate gears *D* and *C*. *B* rotating in an opposite direction to *A*, as shown by the arrows.

This mechanism is termed the tumbler gear, because the gears *D* and *C* are supported in a frame which swings about the axis of either the driving or the driven gear. In the case in hand, the intermediate gears are carried in the frame *E*, which rotates about the axis of the gear *B*. Some means, not shown, must be provided by which the rocker frame may be changed from one position to the other, and locked. Fig. 16 shows the mechanism shifted so that the motions of *A* and *B* are in the same direction.

The tumbler gear has been used as a reversing gear ever since present forms of machine tools were first invented. While it has always

* MACHINERY, December, 1907.

given considerable trouble, it has shown up to disadvantage mostly when applied to the modern machine with positive gear feed, where great power has to be transmitted by it. It is the purpose here to show where this gear may be used to advantage, and also to explain the theory on which the principles of its design are based.

All of us have met with this mechanism in some form or other, and may have formed an unfavorable opinion. The prejudice thus created keeps us from fully appreciating the tumbler gear, even when properly designed, and when used in the right place. It has been placed by many along with the worm drive and the spiral gear as undesirable, and to be avoided unless it is absolutely impossible to get along without it. This opinion has been responsible for the adoption of many combinations used for purposes that rightly belong in the field of the tumbler gear, and many times, in order to avoid using this mechanism, much unnecessary complication has resulted.

What are the faults of the tumbler reversing gear? That one on So-and-So's lathe used to kick furiously when one tried to throw it over. Then, the one used on the milling machine used to go into mesh easily enough, but when any amount of strain was put onto it, the teeth used to crack and growl, showing that the tendency was to drag the gear farther into mesh, causing the teeth to bind on one another and sometimes break. Let us look into the case represented in Fig. 15. Fig. 17 shows the gear *D* just entering into mesh with *A*. An examination of this figure shows that the tendency is for the teeth of gear *A*, when they strike those of gear *D*, to cause the latter to rotate about the axis of the rocker frame, should the gear *B* be locked against turning. This tendency opposes the motion in the opposite direction necessary to bring the gears wholly into mesh. In practice, *B* is not locked, but it is necessary to overcome a certain amount of resistance in order that it may be set in motion, and the presence of this resistance has the same effect as if the gear were locked. The greater this resistance is, the greater is the effort necessary to bring the gears into working position.

Examining the conditions in the case of Fig. 16, we see that the effect would be just the opposite, that is, the gears would come into mesh of their own accord as soon as a contact is produced between the teeth of *A* and *C*. Practically no effort is necessary to bring the gears into mesh, but, in order to withdraw the gear *C* from *A*, considerable effort would be required. When the gears *C* and *A* are in mesh and transmit power, the tendency for gear *C* is to crowd farther into mesh with *A*, which has the effect of binding the teeth. Should the pressure of contact be sufficient, the binding tendency would cause the motion to cease, or would break the teeth. This is one of the points on which many have based their verdict against the tumbler gear, and when designed so that such results are obtained, it is not to be wondered at.

Direction of Tooth Pressure in Ordinary Cut Gears

The first consideration in the design of tumbler gears in any form is that of tooth pressure and its line of application. As all cut gears

used in machine tools are made to the 14½-degree involute system, we will confine ourselves to that system. In this, the force tending to revolve the driven gear is not a tangential force, applied as a tangent to the pitch circle, but is a force applied at an angle of 14½ degrees to the tangent of the pitch circle, this 14½-degree line being termed the line of pressure. In case that there may be some confusion as to the above statement regarding the tangential force and the line of pressure on the teeth, the case is graphically shown in Fig. 18. The tangential force is equal to the twisting moment divided by the radius of the pitch circle. This force is equivalent to that which transmits motion between two disks by friction alone, the diameters of the disks being equal to the pitch circles of the gears. This force is, in the case of a gear, resolved into two component forces. One component acts perpendicular to the tangential force and tends to force the gears apart; the other acts in the direction of the line of tooth pressure shown in Fig. 18. The tooth pressure thus is somewhat less than the

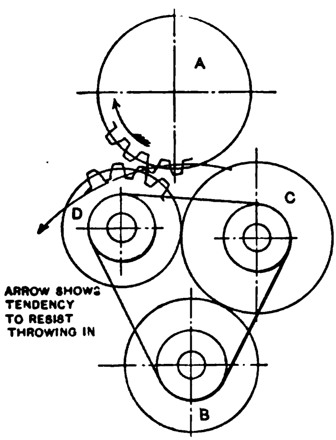


Fig. 17. Illustrating the Tendency to resist Throwing-in of Gear

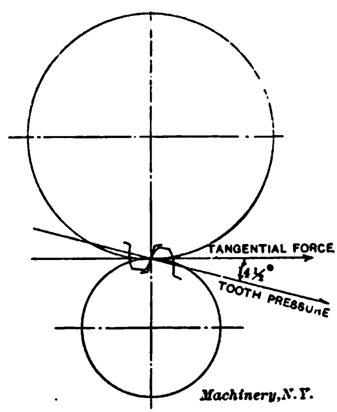


Fig. 18. Illustrating the Direction of Line of Pressure

total twisting force, and equals the twisting or tangential force multiplied by the cosine of 14½ degrees.

Influence of Direction of Tooth Pressure on Tumbler Gear Design

To show what effect the line of pressure has upon the layout of the tumbler gear, we will use the simple case shown in Fig. 19. In this figure, *A* is the driving gear and *B* is the driven gear. These gears are connected by means of the intermediate gear *C*, which is carried in the swing frame *E*, which, in turn, swings about the axis of *A*. This mechanism is a simple case of tumbler gear, and while it is little used, it is useful as a means for disconnecting a train of gears when it is desired to stop the motion of the driven section. If we consider gear *B* locked in the position shown, and exerting a turning effort on the gear *A* in the direction indicated by the arrow, this effort is transmitted by the teeth of *A* and *C*, and a pressure is produced

between the teeth of *B* and *C*, two of which are shown in the cut. The direction in which this force is applied is shown by the line of pressure *H K*, and is is exerted in the direction of *H*. Since every force is opposed by an equal and opposite force when in a state of equilibrium, we have in this instance a force or reaction opposing the force along the line of pressure referred to. It is this reaction that causes our troubles. In the mechanism shown in Fig. 19, the gear *C* and the link *E* are free to rotate about the axis of *A*, and since the line of pressure does not go through the center of gear *A*, the force acting along this line tends to rotate the arm *E* about the axis of *A*, the direction of rotation being dependent on which side of the center of *A* the line falls. Thus in Fig. 19, the line falls in a position that produces a tendency for the arm to force the gear *C* further into mesh with *B*. The twisting moment thus set up is equal to the tooth pressure multiplied by the normal distance from the axis of *A*, or *G L*.

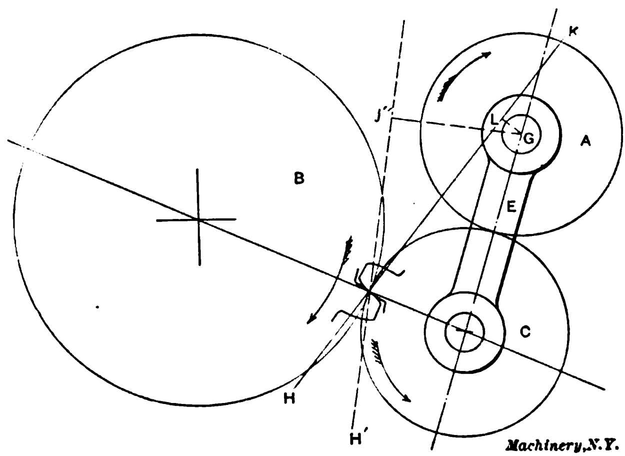


Fig. 19. Objectionable Tumbler Gear Design

If now, instead of trying to turn the gear *A* in the direction of the arrow, we exert a torque in the other direction, the opposite sides of the teeth would come into contact, and the line of pressure would be located as shown by the dotted line *H′ K′*. The normal distance of this line from the axis of *A* is much greater than in the former case; consequently the twisting moment tending to rotate the arm *E* about the axis is also increased, but the direction in which the torque is applied has changed the direction in which the reacting force along the line of pressure acts, and, since this line falls on the same side of the axis, the tendency of the arm is to rotate in the opposite direction, and to separate the gears *C* and *B*. Had the line of pressure gone directly through the axis of the gear *A*, where *E* is pivoted, the effect of any force acting along it would have had no rotating influence upon the tumbler gear arm. That this would be the ideal case needs not to be mentioned, and it should be the aim of the designer to approach that condition as nearly as possible.

The tendency for the tumbler to crowd the gears into mesh might be of some advantage were it desirable to throw them into mesh while the gears are in motion; but in cases where any considerable amount of power is being transmitted, a very stiff and rigid design will be necessary for the tumbler frame and the locking device. It is also well in such cases, when setting the locking device, to have the gears mesh with plenty of play or backlash, so that, if there be any spring in the frame, the gears will not be likely to bind and cramp. Should *B* be the driver and run in the direction of the arrow, the line of pressure would be *H′ K′*, and the pressure would be in the direction of *H′*. The arm would then tend to carry the gear *C* out of mesh with *B*.

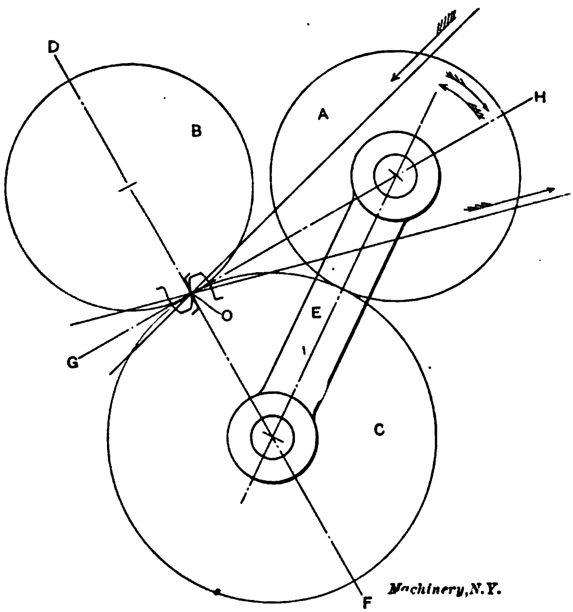


Fig. 20. Correct Design of Tumbler Gear to run in Both Directions

Should the direction be reversed, *H K* would be the line of pressure, and the tendency would be to crowd the gear in.

The layout in Fig. 19 has two bad features. In the first place, the gears have a tendency to crowd farther into mesh, which limits the amount of power that can be transmitted, and increases the liability of breakage of the gear teeth and of the tumbler frame, should an overload be imposed upon the mechanism. Inaccuracy in the shape and spacing of the teeth aggravate the above conditions. In the second place, the mechanism should be used to transmit motion in but one direction.

In most cases the throwing in or out of the tumbler is a secondary matter, as it is done either when the gears are not in motion, or while not under load, if running. In such cases it should be the aim of the designer to overcome the objection of the crowding of the teeth into

mesh by having the line of pressure properly located, so that the tendency is in the opposite direction. When it is desired to provide a tumbler gear that can be run in either direction, the layout in Fig. 20 is recommended. The object in this case is to have the twisting moment equal in either direction, and such that the gears have no crowding tendency. The arrangement in Fig. 20 is laid out as follows: Draw the pitch circles of the gears *B* and *C* and connect their centers by the line *D F*. Through the pitch point *O* draw a line *G H* normal to *D F*. Then locate the gear *A* at some point on *G H* so that its pitch circle will be tangent with that of *C*.

The Single Tumbler Gear

The single tumbler gear is the basis of many of our modern rapid change speed and feed mechanisms, and the principles treated above apply to this as well as to the regular tumbler gear. Take the simple case shown in Fig. 21, which shows the pitch circles of a four-gear

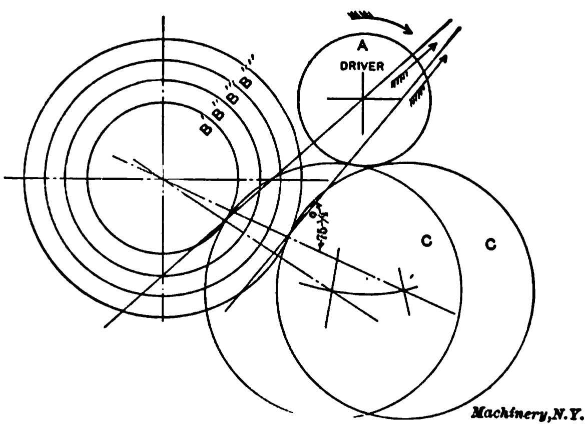


Fig. 21. Single Tumbler Gear in a Feed Box

cone and the driver *A* and tumbler gear *C*. It is evident that only one position of the gear *C* can be such that the ideal condition prevails, that is, only when in mesh with one gear of the cone can the line of pressure pass through the axis of the tumbler frame. Fig. 21 shows this to be the case when *C* is in mesh with the gear *B'*. Each subsequent shifting of the tumbler along the cone brings the line of pressure eccentric to the axis, until the position of extreme eccentricity is reached when *C* is in mesh with *B''''*. In mechanisms of this kind, it should always be the aim of the designer to have the line of pressure pass as close to the center of rotation of the tumbler frame as is possible, because the locking devices used with this type of tumbler gear are necessarily of such a design as to be quick in action, and in consequence are not very stiff or rigid. The line of pressure should always be made to fall on that side of the axis where the tendency is to separate the gears rather than to bring them closer together. When the gear *C* is supported in a swinging frame which does not

slide in a lateral direction, but the changes are made by shifting C along an intermediate shaft, the supporting member should be located at the end where the line of pressure has the greatest eccentricity, as the greatest strain comes at that end. Thus, in Fig. 21 the support should be at the same end as B''''. The diameter of the intermediate gear C has an important effect on the location of the line of pressure. It will be found that it should in most cases be as large as B'''', in order that the line of pressure may come right. However, no exact rule can be given by which the diameter of C can be calculated, as it depends greatly on the difference in the diameters of B' and B'''', and also on the diameter of A.

Rules for the Design of Tumbler Gears

What direct rule can be given that may be used as a guide in laying out the tumbler gear? Referring to Fig. 19, we see that the gear C is revolving in a direction away from the axis of the tumbler at the point of tangency of the pitch circles of C and B, and that the reacting force tends to crowd the gears farther into mesh. Had this line of pressure fallen on the other side of the axis of the tumbler frame, the tendency would have been opposite in effect. When the gear C is revolving so that a point on the pitch circle travels away from the pivot of the tumbler, and the line of pressure falls somewhere between the pivot point and the axis of the driven gear B, the tendency will be to crowd. From this we therefore may formulate the following rules:

Rule I. When the gear about which the tumbler gear swings is the driver, and the line of pressure falls between the axis of that gear and that of the driven gear, the motion of a point on the pitch circle of the tumbler or intermediate gear, when near the contact point, must be toward the axis of the tumbler frame. Should the direction of a point on the pitch circle be opposite, the line of pressure must fall outside of that area included between the axis of the pivot gear and the driven gear.

Referring again to Fig. 19, it is seen that should the driving gear be B, the above rule does not apply, but may be altered to read thus:

Rule II. When the gear about which the tumbler gear swings is the driven gear, and the line of pressure falls in the space between the axis of this gear and that of the driving gear, the motion of a point on the pitch circle of the intermediate gear at the contact point must be away from the axis of the pivot gear; when the line of pressure falls outside of this space, this motion must be reversed.

By following these two rules, more as a precaution than as a compulsory condition, much better success may be expected in the results obtained.

CHAPTER V

FAULTS OF IRON CASTINGS*

POINTS FOR THE MACHINE TOOL DESIGNER

The most useful and indispensable of all the materials with which the designer has to do, is cast iron. Of all the metals used in the construction of machinery, it is the cheapest. It is the one to which we can the most readily give the form and proportions which we desire. It is, of all the common materials, the most easy to machine. While it is lacking in strength and ductility, its cheapness makes it possible to use it in such ample quantity as to overcome these disadvantages, and in many constructions it may be so shaped and proportioned, or so reinforced by other materials, as to make this lack but a slight detriment. It is therefore a matter of interest to the designer to learn of the various faults to which this valuable material is subject, and the best ways in which they can be avoided or minimized.

Causes of Blow-holes

Probably the one fault which spoils more castings than any other, is the result of an outrush of gas from the materials of the cores or the mold, into the molten iron, at the instant of solidification. If the solidification of the iron has proceeded so far that the outrushing gas or steam cannot bubble through it, and escape through the vents which should be provided for the purpose, it will be imprisoned in the substance of the casting, forming one or more holes, according to the special shape of the casting, and the quantity of the escaping gas. These holes, which are known as blow-holes, may not be apparent on the outside, and quite often occur in such a location that they do no particular harm, but it is more often the case that they occur at some point where they become apparent when the metal is being cleaned, or where their presence weakens the casting greatly.

Steam from Moisture in Sand

The gases which cause blow-holes may come from three sources. They may be, and generally are, caused by the generation of quantities of steam from the moisture contained in the molding sand, by the heat of the iron. In the case of dry sand and loam castings, the quantity of steam so generated is so insignificant, if the molds have been properly heated, that it gives no trouble whatever. In the case of green sand castings, however, the moisture present, and therefore the steam generated, is quite large in amount, and special precautions have to be taken to prevent blow-holes.

When the molten iron pours into a green sand mold, all the moisture in the layer of sand immediately in contact with the iron will at once be transformed into steam. The depth of the sand layer so affected

* MACHINERY, October and November, 1907.

depends on the thickness and extent of the fiery mass to which it is adjacent. The steam so formed must either force its way through the molten iron in the form of a mass of bubbles, or else it must escape through the sand. To facilitate its escape, the mold is vented. That is, after the damp sand has been packed around the wooden pattern by ramming it closely into place, a wire is thrust repeatedly into the mold, making numerous passages for the escape of the steam and gas.

It is obviously impossible that one of these vent-holes should extend to every point in the layer of sand adjacent to the casting, so it is necessary that most of the steam and gas should force its way for some small distance through the sand, before it can reach a vent-hole. This it can only do when the sand is somewhat porous. If the sand is too tightly rammed, it will lack the necessary porosity, and even though it be unusually dry, and the venting carefully done, the casting will be full of blow-holes. Cases have occurred where molds have been rammed so hard that the castings were nothing better than shells, the whole interior being a mass of blow-holes.

Decomposition of Binder in Cores, and Entrapping of Air

The second cause of blow-holes in iron castings is the decomposition of the material, generally flour or molasses, used as a binder in preparing the cores, and its escape in the form of gas, into the iron, at the instant of pouring. It is impossible to prepare and bake a core in such a way that it will not give off large quantities of gas when the iron is poured, and so means must be provided for the escape of this gas. In order to do this, the cores are prepared with wax strips running through them. When the core is baked, the wax melts, leaving passages, known as core vents, for the escape of these gases. If the core is of such form, and so set in the mold, that the gases can escape from these vents in an upward or sidewise direction, and leave the mold without forcing their way through the molten iron, no blow-holes will result.

A third source of blow-holes is the entrapping of air in certain parts of the mold, and its mixing, on expansion, with the iron. This trouble is due to insufficient venting of the mold, and is not a fault to which the designer need pay any particular attention.

Dry Sand or Loam Advisable for Large Complicated Castings

In the case of large and complicated castings, it is generally advisable to make dry sand or loam castings, in order to avoid, as far as possible the chance of blow-holes. When the mold is very large, it is difficult to vent it thoroughly, and when the work on it extends over a period of three or four weeks, it is impossible to keep the vents from filling up; hence the general use of dry sand work for large castings. Often, however, for the sake of economy, fairly large and complicated pieces must be undertaken in green sand, and it becomes a matter of importance that they be so designed that the molder will not be compelled to invite disaster by keeping his sand too wet, or ramming it too hard, and that there be no part of the mold which may not be thoroughly vented.

Elements of Green Sand Molding

In order that we may understand thoroughly the effect of the design of a casting on the probability of blow-holes, it is necessary that we review, in a brief way, the elements of green sand molding. The sand is sprinkled with water, and thoroughly mixed and sifted, preparatory to packing or "ramming" it around the pattern. The object of wetting the sand is of course to cause it to stick when it is packed. Up to a certain point, the wetter it is, the better it will stick, but the molder should not wet it any more than is necessary. In the same way, the more tightly the sand is rammed, the better its particles will cohere, and the more easily will the mold be handled, and the pattern drawn. However, tight ramming and wet sand, while they make a solid and easily handled mold, invariably produce blow holes, and are therefore to be avoided.

It will be apparent then, that if a pattern be of complicated form, or hard to draw, or if when it is drawn it leaves the sand in such a form that the mold will easily fall together at a little jarring, the molder will be compelled to wet the sand more and to ram it harder than usual. Small, deep openings, sharp fillets, and thin walls and partitions of sand, are especially troublesome. Not only do they make it difficult to draw the pattern, and handle the mold, and so make excessive wetting and hard ramming imperative, but they cause spots in the mold which the venting wire is unlikely to reach. For these reasons, they are to be avoided when possible, in any class of molding, whether it be green sand, dry sand, or loam work, and on no account should such work be permitted in the case of large green sand castings.

When designing a casting to be made in green sand, the designer ought to know the position which it will occupy in the mold, when it is poured. In general, the parts of a casting which lie in the bottom of the mold will be the soundest, and those parts which must be machined, or which require the greatest strength, should therefore occupy the bottom of the mold, if possible, when the casting is poured. Having decided which side will be down, the designer needs generally to pay no particular attention to the configuration of the lower part of the mold, provided only that all of the pattern can be drawn, and that there are no sand partitions which overhang, or whose extent is large in proportion to their thickness. To insure a sound casting, the sand in the lower parts of the mold must be comparatively dry, and loosely rammed. This condition of affairs is not generally hard to attain, since all the work on the sand is done with the pattern in place, and that part of the mold is not generally moved or handled after the support of the pattern has been withdrawn. In the lower part of the mold, the sand is generally supported at all points in a very thorough manner by the sand lying under it, and so hard ramming or wet sand is unnecessary. If, however, the pattern must be made with loose pieces, or with sharp fillets, or must leave thin walls or tongues of sand when it is withdrawn, the case is changed. Then hard ramming and wet sand are almost compulsory, and the molder

is not to be blamed if he does not produce sound green sand castings. The fault is with the designer.

The upper part of the mold must of necessity be rammed harder than the lower part, since the sand is not supported from beneath, but hangs from above. This is not as great a disadvantage as it might seem to be at first sight, since the escaping gases do not have to make their way through the iron, as they would if they were given off by the sand in the lower part of the mold. The venting, however, must be just as thorough, and it is desirable that the sand should be as dry as possible. The whole arrangement of the upper part of the casting should be such that the sand may be well supported from above. Generously rounded fillets and corners, simple surfaces, plenty of "draft," and an absence of depending walls and masses of sand, make the mold easy to handle, and therefore promote freedom from blow-holes.

When Green Sand and Dry Sand Both May Be Used

It often occurs that the larger part of a casting is of simple form, and easy to mold. A certain part of it, however, may be of a form exceedingly difficult to mold, and therefore likely to give a great deal of trouble. It is not necessary that the whole casting should be made in a dry sand mold, but a core-box may be made to take care of the difficult part of the work, even though the work would ordinarily be done without a core. It is just as easy, and often just as desirable to cast the external face of a casting against a core, as the internal face. While it may not pay to do this if only one casting is wanted, if a great many are wanted it is often the cheapest possible way of making them, and reduces to a minimum both the work of the molder and the chance of a spoiled casting. Often forms may be cast in this way which could not be attempted in any other. If it is desirable to use this method of working, the designer has it in his power to make the construction of the core-box much simpler and cheaper than it might otherwise be, by giving the matter a little thought.

In arranging the coring of a mold, it is always better, if possible, to support the cores at the top. The gases formed in the core, being light, tend to rise, and if the core is supported at the bottom only, they tend to escape into the iron, and to bubble through it. If they can escape at the top, they will pass off without coming in contact with the iron. When it is impossible to support the cores at the top, they should be so arranged that the gases can pass off at the sides, and escape from the mold without coming in contact with the iron.

Sponginess

A second fault to which iron castings are subject is that of sponginess. Sponginess is due to the formation of gas bubbles in iron, at the instant of solidification. In all ordinary cases this is due to an improper mixture of iron. However, if the casting is very thick at one place, and thin at most others, it will be impossible to obtain a mixture which will have satisfactory properties for general work, and not be spongy at points of extraordinary thickness. It is an excellent

rule to allow no part of a casting to be at a greater distance from a sand surface than 2½ inches. In case this rule is strictly adhered to, and the castings are of fairly uniform thickness no trouble will be experienced from sponginess, except from the use of poor iron. When, however, we are obliged to concentrate a considerable quantity of metal at one place, and give it a greater thickness than 5 or 6 inches, either we must take care that it will be at some point where the sponginess will do no harm, or else we must make provision to do away with it.

The only practical method for doing this is to place a riser immediately over the heavy spot. When the metal is poured, and the riser is full, a rod of wrought iron is inserted and worked up and down until the metal has almost solidified. By so doing, the bubbles have a better chance to escape, and the iron is left perfectly solid. Of course, it is not possible to use a riser effectively in this manner, unless it can be placed directly over the heavy spot. A riser at a point a few inches distant is useless. The use of a riser in this way, and for this purpose, is unnecessary when the part of the casting in which the heavy spot occurs is subject to no particular stress, or is not required to be tight under steam, air, or hydraulic pressure, but nevertheless, a spongy spot is a defect in a casting, which should, if possible, be avoided.

Shrink-holes

A third fault to which iron castings are subject is that of shrink-holes. A shrink-hole is a cavity in a casting caused by the shrinking away of the metal in cooling. Like sponginess, this defect is especially likely to occur in those parts of a casting which are excessively thick. To avoid this fault, it is best to avoid sudden changes in the thickness of a section. If the part of a casting which is unusually thick does not have to be machined, the difficulty may be overcome by placing in the mold at that point a piece of iron, so that the casting will be caused to solidify at that point first, on account of the chilling effect of the cold iron. If, however, the heavy spot in the casting has to be machined, or if it is subject to heavy stress, this method of preventing shrink-holes is to be avoided, since the chilling of the iron makes it so hard as to be impossible to cut with a tool, and at the same time creates stresses within the metal which weaken it. In such a case, shrink-holes are best prevented in the same manner as has already been described for the prevention of sponginess, namely, the use of a heavy riser, and the working of the iron with a rod when it is cooling.

The designer must therefore avoid heavy spots in castings, whenever possible, for the reason that they are likely to produce two serious faults, sponginess and shrink-holes. He must avoid them especially in those parts of castings which are to be machined or which are subject to heavy stresses. If they cannot be avoided entirely, in such a case, they should be so arranged that risers may be placed immediately over them, so that a rod may be inserted into the riser, and into the heart of the spot where the metal is thickest.

Scabbiness

A fourth fault often encountered in iron castings is that of scabbiness. Although iron in the molten condition does not permeate the sand of the mold, as water would if it were poured in, nevertheless, on account of the great weight of the fluid, it has a considerable erosive action on the materials of the mold. If, as it flows into the mold, the iron eats away fillets or partitions, or scours away patches of sand, it is obvious that the casting will not be of the proper form, but will have its angles partly filled up, and unsightly protuberances upon its surfaces. Such imperfections as these are known as scabs. The sand so washed from its proper place may float on the iron, and rise to the top of the mold, where it forms a dirty mixture, which, when cleaned away, leaves a rough depression in the surface of the casting, also known as a scab.

The remedy for this trouble is to avoid as far as possible sharp fillets, and thin tongues of sand, projecting into the mold, and to so gate the casting that the current of iron, as it enters the mold, will spread itself out, and not concentrate itself in any particular direction, for if it does, it will eat away the part against which it flows, just as quickly and surely as would a current of water. In general, proper gating is a matter which must be attended to by the molder, but if the designer has arranged things so that proper gating is inconvenient or impossible, the castings will almost surely be scabby.

Sand-holes

The fifth fault to which iron castings are subject, namely sand-holes, is one which is almost invariably associated with that of scabbiness. If the sand which has been eroded by the entering current of iron does not rise immediately to the surface, the iron may partially solidify before it will float to the top. As a result, it will rise till it strikes a part of the iron which has so solidified, and will remain there, imprisoned within the body of the casting. Sand-holes generally occur in that part of a casting which lies near the top of the mold, and just a little ways under the skin. They may occasionally form large cavities which seriously impair the strength of the casting, but more often they form very small holes, which, being full of sand, destroy the edge of any tool which may be used for the purpose of machining the casting, and leave the finished surface pitted and unsightly.

From this description of their origin, it must be apparent that the cure for sand-holes, as far as the designer's work is concerned, is the same as that for scabbiness. The fault may also be avoided by the use of a riser, so arranged that the current of iron will sweep the loose sand out of the mold and into the riser, where it will do no harm; but while this remedy eliminates sand-holes, it does nothing to remedy scabbiness, which is generally the cause of sand-holes.

Floating Cores

A sixth difficulty often encountered in the production of sound iron castings, is caused by floating cores. The buoyant effect of the molten

iron on a core is equal to about three times the weight of the core, if it is solid, and very much more than that in case the core is hollow. Large cores are generally built-up about cast-iron skeletons known as core frames. These core frames are roughly of the same shape as the core, and serve to support it, and to bind it together. Were it not for these frames, heavy cores would fall to pieces by their own weight, and would be broken up in the process of casting, owing to the buoyant effect of the iron. A projecting piece of core having a volume of four cubic feet, for instance, will weigh approximately 500 pounds, and have a buoyant force of about 1,500 pounds thrusting it upward when the mold is poured. If the core frame is not amply strong and stiff, this force will bend the projection upward, or even break it off entirely. Hence the necessity of making large cored cavities of such form that the cores may be rigidly anchored and thoroughly secured. Nor is it sufficient that provision be made for securing the cores, but they must be of such shape, and so reinforced by the frames, that they will be stiff and strong, and not bend appreciably under the tremendous forces which will act on them when they are surrounded by the molten iron.

One of the most difficult things to cast properly is a long iron pipe having a small diameter and thin walls. If such a pipe be cast in the usual position, that is, lying horizontally, there will be an upward thrust along the whole length of the core, tending to bend it. On account of its slenderness, the central portions will be deflected upward, making the walls of the pipe thinner on one side, and thicker on the other. Often the deflection proves sufficient to thrust the core against the side of the mold, if it is long, or, in case the thickness of the wall is great in proportion to the length of the core, to break it off entirely. On this account, pipes and hollow columns of cast iron are often cast on end, thus avoiding any deflection of the core. The same principle may be applied to many other pieces, by taking care to so design them that long and slender cores shall have a vertical position when the mold is poured. If they have such a vertical position, and are supported at both ends, they will have no tendency to deflect one way or another, and this source of trouble may be completely avoided.

Cold Shuts

The seventh fault is known as a cold shut. A cold shut is caused by the imperfect uniting of two or more streams of molten iron, flowing together, which are too cold to coalesce. Such a fault often occurs on the upper side of a thin cylinder cast horizontally, when the iron is not sufficiently hot at the instant of pouring. It there appears as a seam in the side of the cylinder, where it is very apparent that the metal has united imperfectly. It is not only a weak spot in the casting, along which it will readily split if called upon to sustain any great stress, but it is a spot which will surely leak under pressure, and which it is impossible to calk. The cause of the imperfection is generally improper gating, or else too great thinness of metal. If the iron is obliged to flow in thin streams for long distances, it will be

cooled very much, and probably the advancing face will be partially solidified. Consequently, when it meets a similar advancing face of metal, which has been similarly cooled, there is small likelihood of their uniting properly.

The remedy is obviously to so design the casting that the metal will not have to flow in thin streams for long distances. The arrangement of gates and risers is often of great importance in minimizing cold shuts, and if the casting is large, and at the same time has thin walls, the designer must see that the gates may be so arranged that the iron may quickly fill up the mold. While the arrangement of the gating generally depends on the molder's fancy, he may often be limited by the shape of the casting, and obliged to place the gate at some point where the iron, in flowing in, must spread itself into a thin sheet, or pass for a considerable distance through a narrow passage. Under such circumstances, a cold shut is hardly to be avoided.

Shrinkage Strains

The eighth and last fault is that of shrinkage strain. If we have two pieces of iron fastened end to end, as shown in Fig. 22, one piece being notably thinner than the other, the thinner piece will solidify first in the mold, and cool some hundreds of degrees below its freezing

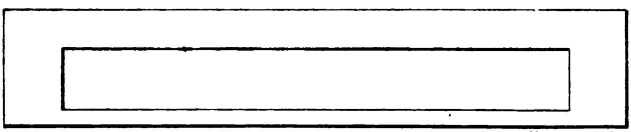

Machinery, N. Y.

Fig. 22. Casting of such Shape as to be Subjected to Severe Internal Stresses

point, before the thicker part solidifies. As a result, the thicker part, when cooled to air temperature, will have, or rather tend to have, a less length than the thinner part, the reason being that at the instant of solidification of the thicker part, both pieces had the same length, although the thinner part was much the cooler. The thin part will then be in compression, while the thick part is in tension, and severe stresses will exist within the piece, which make it weaker than it would otherwise be in most cases.

Sometimes, however, we are enabled to utilize the shrinkage stresses to advantage. For instance, when cast iron was the standard material for the manufacture of ordnance, guns were cast with cores through which water was circulated, so as to cool the surface of the bore before the outer parts solidified. When a gun is fired, it is known that the inner layers of metal are stretched more than the outer ones. By cooling the inner layers of metal first, shrinkage strains are produced in the walls of the gun, causing the outer layers of metal to compress the inner ones. The combined effect of the shrinkage stresses and the stresses produced by the explosion is to produce a uniform stress throughout the walls of the guns, and so reduce the chance of rupture.

It is not often, however, that we are able to take advantage of shrinkage strains in this way. More often they are troublesome, caus-

ing work to warp in the process of machining, or causing mysterious cracks to develop without apparent cause. Since these strains are due to unequal rates of cooling in the different parts of the casting, the best way to eliminate them is to so arrange the thickness of the various parts, that the entire casting shall solidify at the same time. The second best way is to so arrange the parts of the casting that the unequal contraction shall not produce dangerous stresses at any point. In order that the entire casting shall cool at a uniform rate, it is necessary that all parts of it shall be of approximately uniform thickness, and that there shall be no sudden changes of section. In order that unequal contraction shall not produce dangerous stresses in the metal, it is necessary that there shall be no sharp corners, and that the various parts shall be free to expand when necessary. For instance, a wheel or pulley with a solid rim is likely to have severe stresses

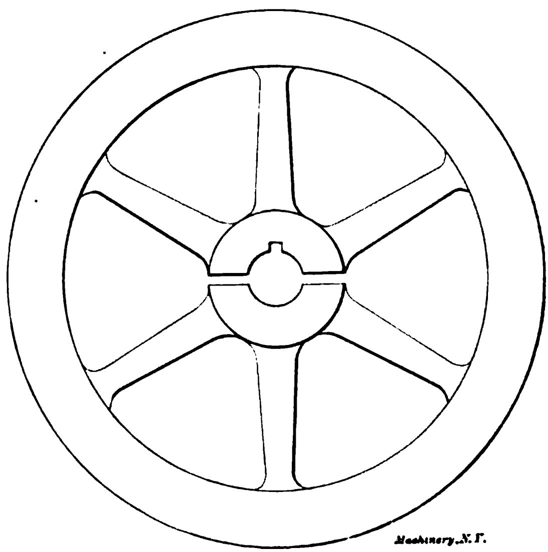


Fig. 23. Method of Obviating Shrinkage Strains in Large Wheels

set up within the arms by unequal cooling, but if the hub be divided as shown in Fig. 23, by means of a thin core, and then bolted subsequently, no shrinkage strains will occur, since the arms are free to expand or contract, independently of the rim.

Shrinkage strains often become so serious that it becomes necessary to make pieces in two or more parts, which it would be perfectly possible to make, at much less expense, in one piece. Large jacketed cylinders, for steam and gas engines, are good examples of this. When cast in one piece, the shrinkage stresses, together with the stresses set up by the varying temperatures incident to services, are often sufficient to crack them. Were the piece shown in Fig. 22 made in two parts, as shown in Fig. 24, there would be no shrinkage strains in either part, although the cost of machining the surfaces which are

fitted together, and of putting in the bolts, would not always warrant the construction.

Relative Economy of Simple and Complicated Castings

In conclusion, it may be well to state that most of the faults enumerated will be more likely to occur in a part of a complicated casting, than in a similar part of a simpler casting. For instance, the cylinder of a gas engine will be more likely to have some imperfection if it is cast integral with the frame, than if it is cast separately.

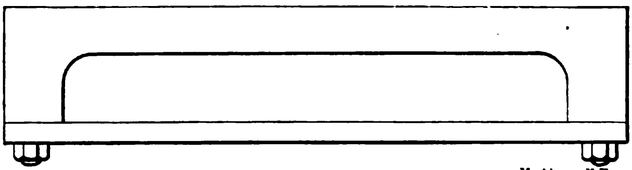


Fig. 24. Piece shown in Fig. 22, made in Two Parts

In the same way, the frame will be more likely to have an imperfection of some kind, than if it were cast separately. Assuming that ten per cent of the cylinders or frames would be lost if they were cast separately, it is more than likely that fifteen per cent of the castings, having cylinder and frame cast together, would be rejected for faults in the frame, and fifteen per cent of the remainder would be rejected for faults in the cylinder. In other words, twenty-eight per cent of these castings would be rejected, against ten per cent of the simpler forms. If more than eighteen per cent of the cost of the castings is saved in machining, or in other ways, by casting cylinder and frame together, it is well to do so, but if the saving is not more than sufficient to balance the loss, it is well to make several simple forms, instead of one complicated one.

CHAPTER VI

PROPORTIONS OF MACHINES BUILT IN A SERIES OF SIZES*

The problem of cost reduction forces itself, with increasing vividness upon the mind of every person who has to do with the manufacture of machinery. To the "small shop" people, and to those whose product is unsystematized and whose ideas of methods to pursue are, as yet, vague, this chapter may prove of some assistance.

There are three important means by which the shop product may be systematized: By the use of formulas; by the use of tables; and by the use of charts. As the two latter may be considered as the tabulated, or graphic results of the former, we will deal only with the formulas. In determining sizes, weights, and costs, these formulas are generally most efficient time-savers. For convenience, formulas in this chapter will be divided into two classes: The class used to produce the first of a type of machine we will call fundamental; and the class used to produce several sizes of this type of machine, empirical. Upon seeking fundamental formulas in text books and in mechanical engineers' pocket-books we are confronted by a diversity of opinions and tabulated results that are, at least to a novice, a bit confusing. These formulas, it is always to be remembered, have their application in the special case under consideration, and are to be used only as guide posts in our journey of design. It is evident to most designers that some kind of a tentative method must, sooner or later, be resorted to in the type design, for in nearly all machines the governing conditions soon become so numerous or indefinite as to render a subdivision of the problem a necessity. A certain amount of judgment is absolutely essential in the use of most fundamental formulas, and discrimination is always necessary.

Graphically, fundamental formulas can be represented by curves, and will be correct for all sizes under identical conditions, while empirical formulas rest on no such basis and hold true for but a certain series within certain limits. This constitutes the vital difference between fundamental and empirical formulas. A fundamental formula is one found through mathematical reasoning, while an empirical formula is made up by means of trial methods.

Suppose that we have built two or three sizes of a certain type of engine and that they are successful; we desire to put on the market an entire line. Our sizes of this type of engine will run from 10-inch cylinder diameter in the smallest to 30-inch cylinder diameter in the largest. We have built a 12-inch and a 24-inch engine and perhaps an 18-inch. These engines were, as was imperative, tentatively designed. In seeking the derivation of the empirical formula for the

* MACHINERY, November, 1902.

length of the cross-head shoe, we find that on our 12-inch engine we have given it an area of 55⅛ square inches, and on our 24 inch engine its area is 190⅛ square inches. In each case the length of the shoe was nearly twice its width, so we decide to make it so in our line of engines; solving for the width, we have in the 24-inch engine

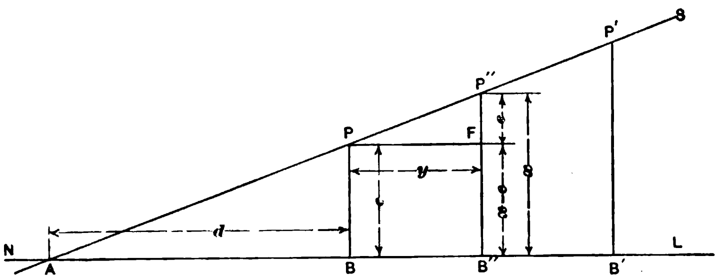
$$2x^2 = 190.125; \quad x = \sqrt{95.0625} = 9.75.$$

making our shoe length for the 24-inch engine 19½ inches, and for the 12-inch engine 10½ inches.

To any scale in Fig. 25, perpendicular to the line NL, lay off these shoe lengths PB and $P'B'$—10½ inches, and 19½ inches, respectively—making the distance BB' equal to 12 inches, the difference between our sizes 12 inches and 24 inches. Through points PP' draw line SA intersecting NL at A. At B''—for our 18-inch size—erect a perpendicular $B''P''$. Draw PF intersecting $B''P''$ at F. Using the notation given in the figure, we get the simple equations

$$e:y = c:d; \qquad ed = cy; \qquad e = \frac{c}{d}y;$$

$$x - c = e; \quad x - c = \frac{c}{d}y; \quad x = \frac{c}{d}y + c.$$



Machinery, N.Y.

Fig. 25

In this last formula many will recognize an old acquaintance—the equation for a straight line. Let us now analyze this equation. From the figures it is seen that x = the desired dimension and that $\frac{c}{d}$ = the rate of increase in the slope of the line. If now we measure the distances and substitute their values for c and d we may determine the ratio $\frac{c}{d}$, which we will call f.

Then $f = \frac{c}{d} = \frac{10.5}{14} = \frac{3}{4}$, and

$$x = fy + c, \text{ or } x = \tfrac{3}{4}y + c.$$

In interpreting our empirical formula $x = fy + c$, we have
y = a common unit to which all other sizes are to be referred,
x = desired dimension,

f = a factor of y,

c = a constant increment to be added in each case.

The unit of value y, as generally selected, is a bolt or cylinder diameter, or the capacity of the machine. Obviously, in our line of engines, we select the cylinder diameter D as our value of y, and our unit formula then becomes $\frac{3}{4}D + c$. The value of c is now determined by direct substitution, in the following manner: x being the shoe length, we substitute for it $19\frac{1}{2}$ inches (its length on the 24-inch cross-head); then

$$\tfrac{3}{4} \times 24 + c = 19\tfrac{1}{2};\; c = 19\tfrac{1}{2} - 18 = 1\tfrac{1}{2}.$$

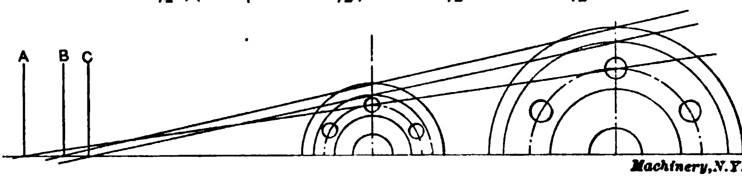


Fig. 26

Note, that while we have assigned to y and c other values, we have not altered the relations; our formula for this particular cross-head dimension now becomes $\frac{3}{4}D + 1\frac{1}{2}$ inches.

For convenience in charting these sizes, some point is determined upon as a pole about which these lines (represented by our formulas) are drawn as vectors, the ordinate length for a particular size giving the desired dimension. If now in the determination of other formulas it be found, as is likely to be the case, that these lines do not all pass

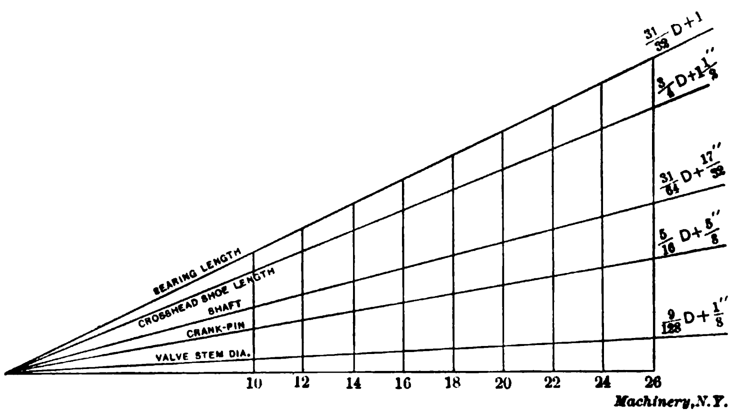


Fig. 27

through a common point, it becomes necessary to select one. In well-designed machines the intersection of these lines with the base line will come close together, and an average of these intersections is selected as a pole. Figs. 26 and 27 will serve to illustrate the purport of this paragraph.

Experienced designers are well aware that the final test of any dimension in a design is that of satisfying all fundamental calculable

conditions; nevertheless, the instances where our empirical formulas prove incorrect are very few indeed. With the design for our line of engines thus systematized, let us consider what are to be the advantages that will naturally result from it. In the first place, the weights of any particular parts, or details of any size in our line of engines may be determined prior to its design or manufacture. In the determination of weights, cubic contents, and similar processes, the use of "differences" as applied to higher mathematics, will not only prove an efficient time-saver, but relieve much of the drudgery attendant upon such operations.

A brief explanation of the use of "differences" is as follows: When we have a series of numbers connected by a regular, though not obvious law, the nature of that law may be discovered by forming a new series of differences between each two terms of the original series, and then

TABLE SHOWING PRINCIPLE OF THE METHOD OF DIFFERENCES

Arithmetic Solution.				Algebraic Solution.				
Col. 4.	Col. 3.	Col. 2.	Col. 1.	Term	Col. 5. Original Ser.	Col. 6. 1st Ser. Differences.	Col. 7. 2d Ser. Dif.	Col. 8. 3rd Dif.
			7	1	W			
		5				y		
	3		12	2	$W+y$		x	
1		8				$y+x$		c
	4		20	3	$W+2y+x$		$x+c$	
1		12				$y+2x+c$		c
	5		32	4	$W+3y+3x+c$		$x+2c$	
1		17				$y+3x+3c$		c
	6		49	5	$W+4y+6x+4c$		$x+3c$	
1		23				$y+4x+6c$		c
	7		72	6	$W+5y+10x+10c$		$x+4c$	
1		30				$y+5x+10c$		c
	8		102	7	$W+6y+15x+20c$		$x+5c$	
1		38				$y+6x+15c$		c
	9		140	8	$W+7y+21x+35c$		$x+6c$	
1		47				$y+7x+21c$		c
	10		187	9	$W+8y+28x+56c$		$x+7c$	
		57				$y+8x+28c$		
	11		244	10	$W+9y+36x+84c$			
		68						
			312					

treating the new series (which we may call the series of first differences) in the same way, until we reach a series of differences, the law of which is obvious. In the table above will be found both the arithmetic and algebraic solutions of problems by "differences."

In column 1 of the table is given a series of numbers, which we suspect follows some definite, though not obvious law, and which we desire to discover. We here take the differences between each two terms in column 1 and put them down in column 2. Having proceeded with the two orders of differences, the law becomes apparent early in the process of determining the values in column 3. Referring again to the table, it is evident that the next term of column 3 must be 11, which gives 68 (57 + 11) as the next term of column 2 (the series of first differences) and 312 (244 + 68) for the original series. Note that this series can thus be obtained indefinitely, and that ultimately,

in any regular series, some one series of differences will become a constant. It is on the principle of differences that calculating machines are constructed to compute logarithmic tables, etc.

In the algebraic solution of such problems as involve the determination of weights and volumes, it will be necessary to calculate these weights or volumes for the 1st, 2d, 3d, and 4th terms of our given series. By substiting in the formulas in column 5 the numerical value of W, which is the first term in our given series, we may equate these expressions and our calculated values for the 2d, 3d, and 4th terms, and determine, by simple algebraic processes, the values of c, x, y, and ultimately, those values which we are requiring in the original series, column 5.

The computations concerning the cost of materials logically follow the determination of volumes and weights and are made with comparative ease. However, our next problem concerning the determination of the cost of labor is a more difficult one to solve. Formulas should express this cost in so many cents per pound of product, including all shop charges, and be established partially by experience and partially by methods suggested in this chapter.

In many instances it will be found both desirable and convenient to have this cost formula embody the unit dimension. When this is the case, the formula is, as are most cost formulas, established by the tentative methods to which we have just alluded. As the methods employed in the deduction of these formulas render them purely empirical, one or another form of expression may have to be adopted. However, formulas of this class usually assume the form of, or at least may be solved into, the familiar type form

$$ax^2 - bx + c,$$

where a and b are factors of the unit dimension, and c is a constant.

For the purposes of illustration we will assume that the formula for the cost of labor which we have established is

$$\frac{3}{2}D^2 - 15D + 314.$$

In this form the formula gives the total cost of labor in dollars for the size desired. The cost of labor C for our 18-inch engine would then be

$$C = \frac{3}{2}(18)^2 - 15 \times 18 + 314 = 530.$$

The computation of a final cost formula, embodying the unit dimension, is the last process in our development of shop formulas; this formula is derived directly from those relative to the costs of material and labor.

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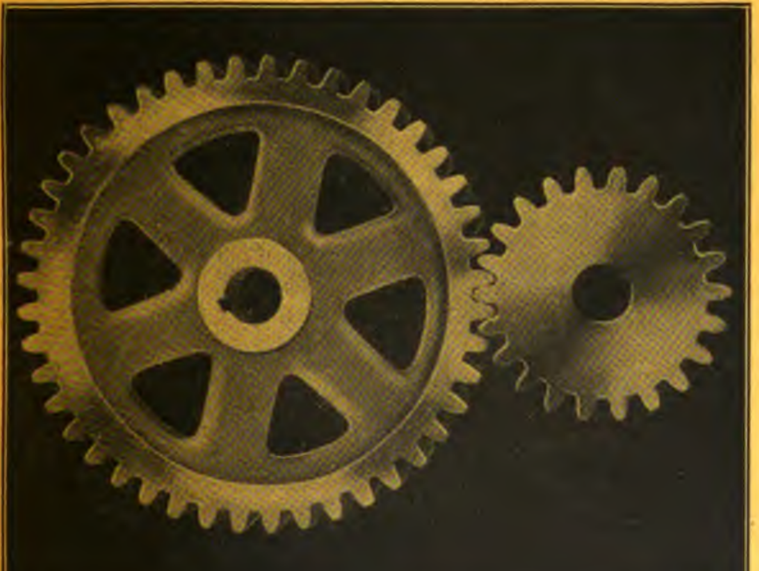

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SPUR GEARING

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CONTENTS

CHAPTER I

FIRST PRINCIPLES OF GEARING*

Gear wheels are such common objects about the machine shop, and are manufactured with such rapidity and ease by the aid of the modern automatic gear cutter, that many seldom stop to think what they really are, why the teeth must be constructed with certain curves, and what it is desired that they shall accomplish. In following chapters we shall take up some of the practical questions, touching upon the calculations that come up in the design, but will here deal chiefly with a few of the theoretical points of the subject that are seldom explained in a simple manner for the benefit of those who have had neither the time nor the opportunity to look into matters of this kind.

Suppose there are two wheels arranged as in Fig. 1 with their faces in close, frictional contact, and that both are exactly the same size, so

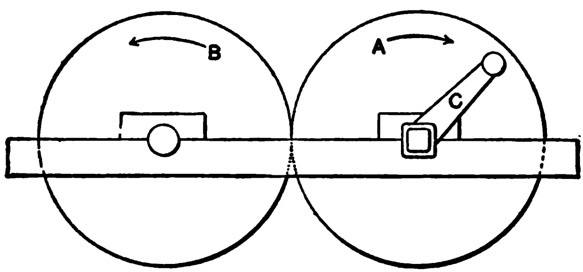


Fig. 1

that when the crank is turned around once, wheel *B* will turn exactly once also, provided, of course, there is no slipping between the two wheels. It must be noticed, moreover, that if the crank be turned uniformly, wheel *B* will not only make the correct number of revolutions relative to *A*, but it will revolve uniformly, as well; that is, both its total motion and the motion from point to point will be correct.

Now there are many places in machine construction where the slipping inseparable from friction wheels cannot be tolerated, and this difficulty might be overcome by fastening small projections to one of the wheels, as on *A* in Fig. 2, and cutting grooves in the other wheel, *B*. Then, if the crank were turned, wheel *B* would always make just the right number of turns, even if considerable power were transmitted. It is probable, however, that these projections and grooves would not fulfill the purpose of gear teeth. What is wanted of gear teeth is that they shall give exactly the same kind of motion as corresponding friction wheels, running without slipping. They must not only keep

* MACHINERY, June, 1898.

the number of revolutions right, but they must give a perfectly even and smooth motion from point to point or from tooth to tooth.

Fig. 3 will show clearly how such a result is obtained. It represents the friction wheels with teeth fastened to them, the teeth, of course, extending all the way around instead of part way as shown. These teeth are set so as to be partly without and partly within the edges of the two wheels, as obviously they will give better results thus arranged than with all the projections on one wheel and all the grooves or depressions on the other, as in Fig. 2.

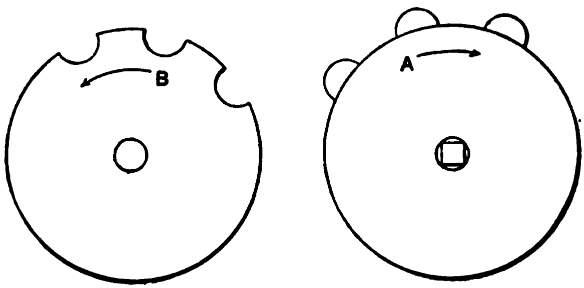


Fig. 2

With the wheels fitted in this way it can be proved that the only conditions which must be fulfilled in order that the teeth shall give wheel *B* the same motion that it would have if it were driven by frictional contact with wheel *A* is that a line drawn from the point *O*, where the two wheels meet, to the point where the tooth curves touch, shall be at right angles to both tooth curves at this point, whatever the

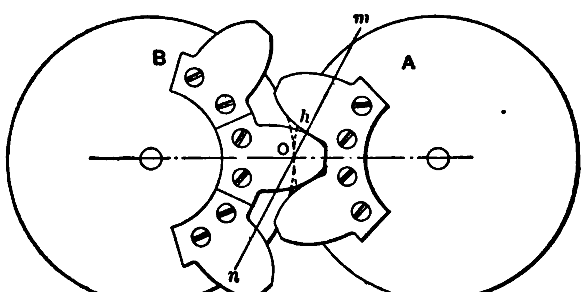


Fig. 3

position of the gears. For example, in Fig. 3, two of the teeth touch at *h*. If the curves are of the right shape, a line *m n*, drawn through *h* and *O*, will be at right angles to both curves at point *h*. This is the law of tooth curves, and it makes no difference what the shape of the teeth is, so far as their correct action is concerned, if this law holds true for every successive point where the teeth come in contact.

In technical language the "friction wheels" mentioned are known as "pitch cylinders," and they are always represented on a gear drawing by a line—usually a dash and dot line—called the "pitch line." As

teeth are generally proportioned, this line falls nearly, but not quite, midway between the tops and bottoms of the teeth, the inequality being due to the space left at the bottom of the teeth for clearance. The diameter of the pitch cylinder is called the "pitch diameter."

Involute System

We are now ready to consider the particular forms of teeth most often used. The one that is at present most in favor is the involute tooth, the term "involute" being the name of a curve described by the

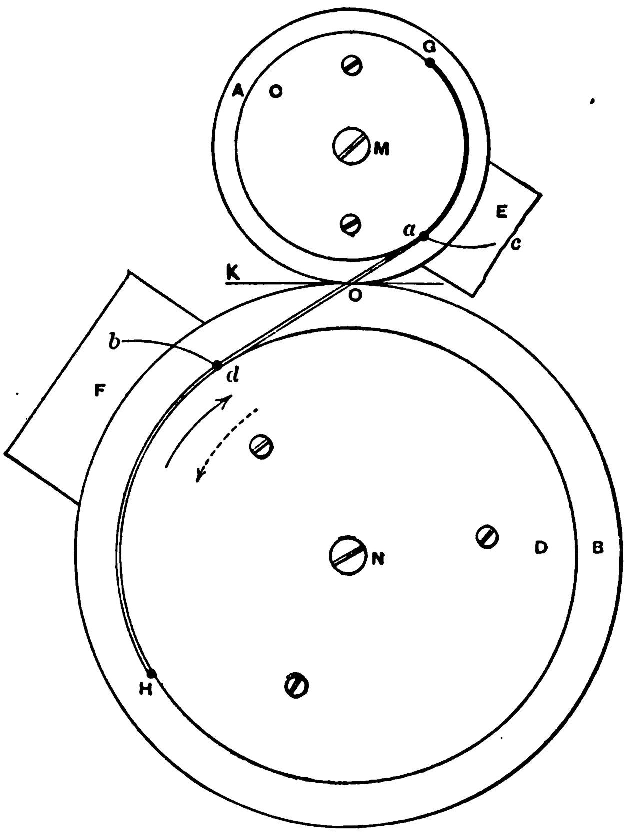


Fig. 4

end of a cord as it is unwound from another curve. For example, to draw an involute, wind a cord around a circular disk of any convenient material, and make a loop in the outer end of the cord. Lay the disk flat on a piece of paper, and with a pencil in the loop, unwind the string, keeping it drawn tight, and let the point of the pencil trace a curve, which will then be an involute.

In Fig. 4 is shown how the same principle is applied to forming tooth curves. *A* and *B*, with centers at *M* and *N*, are two disks which

serve the purpose of pitch cylinders. *C* and *D* are two smaller disks fastened to the larger ones and around which a cord is stretched and fastened at points *G* and *H*. When either disk is turned, the cord is supposed to pull the other one around at the same speed that it would go if moved solely by frictional contact between disks *A* and *B*. To do this, it is simply necessary to have the disks *C* and *D* in the same ratio as *A* and *B*. If *A*, for example, is half as large as *B*, then *C* must be half as large as *D*.

To make room for drawing the curves, let pieces *F* and *E* be fastened to the large and small wheels, respectively. With a pencil fixed at point *d* on the cord, turn the wheels in the direction of the solid

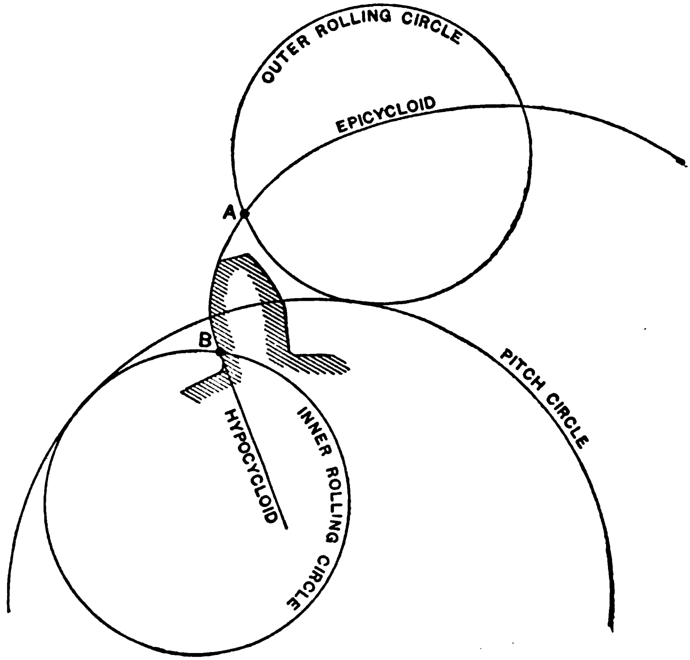


Fig. 5

arrow, meanwhile moving the pencil outward, and the curve *db* will be described, which will be a suitable tooth curve for the larger wheel, and which it can be proved will answer the requirements of the general law. Starting again with the pencil at *a*, and turning the wheels in the direction of the dotted arrow, and moving the pencil outward, a similar curve, *ac*, for the smaller wheel will be traced.

The circles representing the disks *C* and *D* are called "base circles," and in practice are drawn at a distance from the pitch circle of about one-sixtieth of the pitch diameter. This brings the angle, *KOd*, called the angle of obliquity, in Fig. 4, about 14½ degrees; and although it is not by any means certain that this is the best angle, it is the one commonly used.

Cycloidal System

Take a silver dollar and roll it along the edge of a ruler, holding the point of a pencil at the rim of the dollar, so that as the latter rolls, the pencil will trace a curve. This curve is a cycloid. Should the dollar be rolled on the edge of a circular disk, however, the curve traced would be an epi-cycloid, and should it be rolled on the inside of a hoop, it would be called a hypo-cycloid. These curves are employed for the teeth of the cycloidal system of gears.

In Fig. 5 it is shown how the face or the outer portion of the tooth is rolled up by the point *A* on the outer rolling circle, and how the flank or inner portion is generated by point *B* on the inner rolling circle. In this case the hypo-cycloid and flank are straight lines, the reason for this being that, as drawn, the diameter of the rolling circle

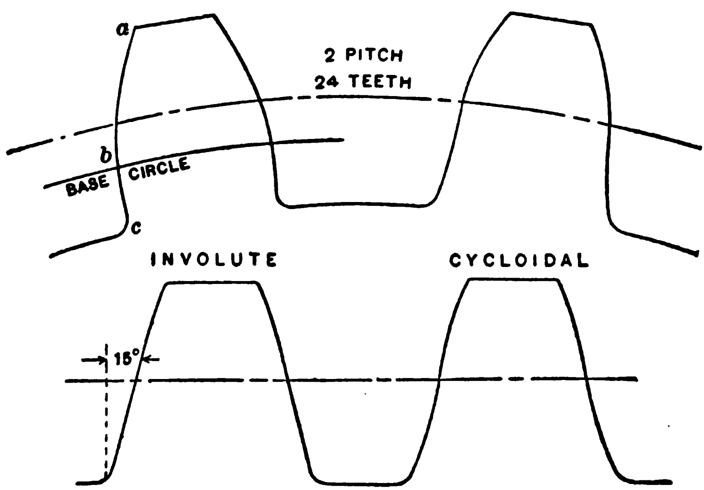


Fig. 6

is one-half the diameter of the pitch circle of the gear, and the hypo-cycloid generated under these conditions becomes a straight line.

Comparison of the Involute and Cycloidal Systems

The involute and cycloidal systems are the only two that are used to any extent, and in Fig. 6 a gear tooth and rack tooth of both are shown for comparison. The involute gear tooth has the involute curve from point *a* to point *b* on the base circle, and from *b* to *c* at the bottom of the tooth the flank is a straight, radial line. One difficulty with the involute system is that with the standard length of tooth the point *a* will interfere when running with gears or pinions having a small number of teeth. To avoid this, the point is rounded off a little below the involute curve. In general appearance the tooth seems to have a broad, strong base, and a continuous curve from *a* to *c*. A strong feature of the involute gearing is that it will run correctly even if the distance between the centers of the wheels is not exactly right. This

will be evident by referring to Fig. 4, where it will appear that the relative velocities of the two wheels will be the same however far apart they may be, and if involute teeth are used in place of the string connection there shown, the action will be just the same. The involute rack tooth has straight sides at an angle of 14½ degrees, with the points rounded off.

Of the cycloidal teeth but little need be said except that they have two distinct curves above and below the pitch line, as previously explained, and that in the rack tooth the two curves are just alike, but reversed.

TABLE I. CUTTERS FOR INVOLUTE GEAR TEETH

No.	1	will	cut	wheels	from	135	teeth	to	a rack.
"	2	"	"	"	"	55	"	"	134 teeth.
"	3	"	"	"	"	35	"	"	54 "
"	4	"	"	"	"	26	"	"	34 "
"	5	"	"	"	"	21	"	"	25 "
"	6	"	"	"	"	17	"	"	20 "
"	7	"	"	"	"	14	"	"	16 "
"	8	"	"	"	"	12	"	"	13 "

Whatever system is used, it is essential that all the wheels of a given pitch should be capable of running together. To make this possible with the involute, all the wheels must have the same angle of obliquity; and with the cycloidal system the same size rolling or describing circle must be employed for all sizes. The circle generally chosen is one having half the diameter of a 12-tooth pinion, which makes the flanks of this pinion radial. In Fig. 5, if the diameter of the rolling circle had been either greater or less than half the diameter

TABLE II. CUTTERS FOR CYCLOIDAL GEAR TEETH

Letter of Cutter	No. of Teeth	Letter of Cutter	No. of Teeth
A	12	M	27 to 29
B	13	N	30 to 33
C	14	O	34 to 37
D	15	P	38 to 42
E	16	Q	43 to 49
F	17	R	50 to 59
G	18	S	60 to 74
H	19	T	75 to 99
I	20	U	100 to 149
J	21 to 22	V	150 to 249
K	23 to 24	W	250 or more
L	25 to 26	X	Rack

of the pitch circle, the flank of the tooth would have been curved, and in the case of the greater circle, the curve would have fallen inside of the radial flank drawn in the figure, causing a weak, under-cut tooth. With the smaller circle, the curve would fall outside, making a strong tooth.

Cutters for Involute and Cycloidal Teeth

According to the system for cutting gear teeth adopted by the Brown & Sharpe Mfg. Co., Providence, R. I., any gear of one pitch will mesh with any other gear or with a rack of the same pitch. Eight cutters are required for each pitch. These eight cutters are adapted to cut from a pinion of twelve teeth to a rack, and are numbered, respectively, 1, 2, 3, etc. The number of teeth and the pitch for which a cutter is adapted is always marked on each. A list of these cutters is given in Table I.

Cutters for the cycloidal form of teeth are also made so that any gear of one pitch will mesh into any other gear or into a rack of the same pitch, but twenty-four cutters are required for each pitch. In order that gears with this form of teeth shall run well together, they must be cut accurately to the required depth; otherwise the pitch circles will not be tangent to each other. To secure a proper depth of tooth, the cutters are made with a shoulder which determines the exact depth that the tooth should be cut. Thus, if care is taken when turning the blanks, to obtain the correct outside diameter of the gear, no measurements need be taken when cutting the teeth. The twenty-four cutters are adapted to cut from a pinion of twelve teeth to a rack, and are designated by letters *A*, *B*, *C*, etc. The number of teeth and the pitch for which the cutter is adapted is always marked on each, the same as in the case of cutters for involute teeth. A list of these cutters is given in Table II.

CHAPTER II

FORMULAS FOR DIMENSIONS OF SPUR GEARS*

When we consider the number of gears used in machinery, and the number of men employed in the manufacture of machines using gears, it is rather surprising to find men who are unable to find the outside diameter, having given the pitch diameter and pitch, or to find the distance between centers of two gears, having given the number of teeth and pitch, and similar problems. The object of this chapter is to explain in as clear and practical a manner as possible the underlying principles of gearing, and to give concise rules or formulas for the solution of problems which arise in our everyday work upon gears.

Pitch Diameters

Two shafts *A* and *A′* (Fig. 7) carry rollers *B* and *B′*. By having pressure on the shafts as indicated by the arrows, and revolving *A*. the friction of the rollers at the point of contact, *X*. will cause *A′* to revolve, but we can readily see that if any great amount of power is to be

* MACHINERY, July, August, October and November, 1897.

transmitted, the rollers are liable to slip at the point of contact *X*, which will not give a positive motion; that is, it will require more than one revolution of the shaft *A* to produce one revolution of the shaft *A'*.

Suppose, as shown in Fig. 8, that we put projections on the surface of the roller *B* and cut recesses in the roller *B'*, making them of such shape that the sides of the projections on roller *B* will slide with as little friction as possible upon the sides of the projections caused by cutting the recesses in roller *B'*. Then, when shaft *A* is revolved, shaft

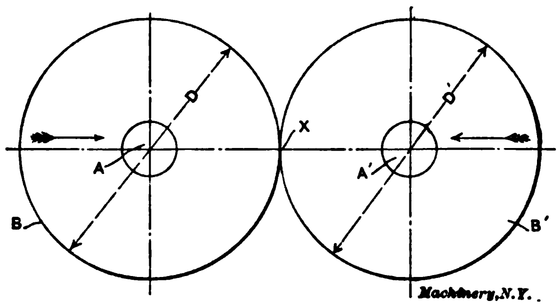


Fig. 7

A' must also revolve. The identity of the rollers *B* and *B'* is not lost, for we have simply added a number of projections to one, and cut the same number of recesses in the other, and the point of contact of the two rollers is still at *X*, but in this case there is no special pressure required to keep the rollers together as in the preceding case, nor is

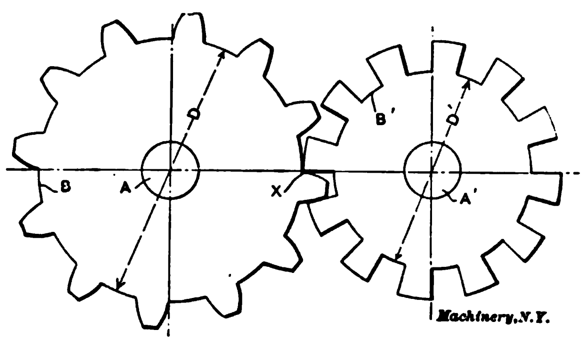


Fig. 8

there any slip, and consequently shaft *A'* will make one revolution in the same time that shaft *A* does.

In Fig. 9 we have changed Fig. 8 by adding projections between recesses in roller *B'*, and by cutting recesses on roller *B* between projections, and we have the regular gear tooth. We have now no visible part of the original rollers *B* and *B'* left, but we have in their places imaginary rollers, the diameters of which are the pitch diameters of the gears. Thus we might have called our original rollers pitch rollers, and then proceeded to put on our projections and cut our recesses,

which would have given us the gear wheel. This has already been explained in a general way in Chapter I.

Of course, in practice gears are never made in this way; the gear blank is first turned up to the correct diameter, and then the space between the teeth is cut. The method of finding the outside diameter will be given later, this illustration being used simply to show the evolution of the gear wheel from the friction disks or pitch rollers.

Pitch

When we speak of the pitch of a gear, the diametral pitch is generally referred to. The gear really has two pitches, diametral and circular. The *diametral pitch* of a gear is the number of teeth for each inch of pitch diameter. If a gear has 20 teeth and the pitch diameter is 2 inches, the diametral pitch would equal 20 ÷ 2, or 10; or there are 10 teeth in the gear for each inch of pitch diameter which it contains, and we would call it a 10-pitch gear. The *circular pitch*

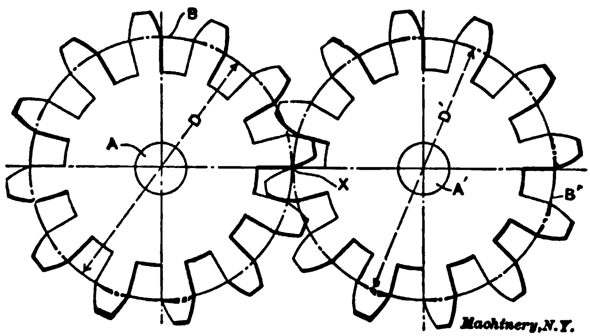


Fig. 9

of a gear is the distance from the center of one tooth to the center of the next adjacent tooth, measured on the pitch lines. It is very seldom that circular pitch is used in describing cut gears.

It can readily be seen that the circular pitch being equal to the distance from the center of one tooth to the center of the next, must be the result of dividing the circumference of the pitch circle by the number of teeth in the gear. Should an occasion arise where it would be necessary to obtain the circular pitch, having the diametral pitch given, divide 3.1416 by the diametral pitch, and the quotient will be the circular pitch, or, expressed in its simplest form,

$$\frac{3.1416}{P} = P' \qquad (1)$$

in which P = diametral pitch; P' = circular pitch.

Example.—If the diametral pitch of a gear is 4, and it is required to find the circular pitch, divide 3.1416 by 4, and the quotient, 0.7854, is the circular pitch of the gear.

If the circular pitch be given, to find the diametral pitch, we can readily see that formula (1) would have to be transposed and would read thus:

$$\frac{3.1416}{P'} = P \qquad (2)$$

P and P' representing the same as before.

Now, having given the rules, we will proceed to explain how they were obtained. We know that the distance around the circumference of a circle is equal to 3.1416, multiplied by the diameter of the circle; consequently, for every inch of diameter we have 3.1416 inches of circumference. If the diametral pitch of a gear is equal to the number of teeth for each inch of pitch diameter, and each inch of diameter is represented by 3.1416 inches of circumference, then the diametral pitch equals number of teeth for each 3.1416 inches of circumference. As the circular pitch is the distance from the center of one tooth to the center of the next, then the circular pitch must be equal to 3.1416 divided by the number of teeth in that 3.1416 inches of circumference, and, as we have shown that the diametral pitch is equal to the number of

TABLE III. DIAMETRAL PITCH CONVERTED INTO CIRCULAR PITCH

Diametral Pitch	Circular Pitch	Diametral Pitch	Circular Pitch
2	1.571 inch.	12	0.262 inch.
2¼	1.396 "	14	0.224 "
2½	1.257 "	16	0.196 "
2¾	1.142 "	18	0.175 "
3	1.047 "	20	0.157 "
3½	0.898 "	22	0.143 "
4	0.785 "	24	0.131 "
5	0.628 "	26	0.121 "
6	0.524 "	28	0.112 "
7	0.449 "	30	0.105 "
8	0.393 "	32	0.098 "
9	0.349 "	36	0.087 "
10	0.314 "	40	0.079 "
11	0.286 "	48	0.065 "

TABLE IV. CIRCULAR PITCH CONVERTED INTO DIAMETRAL PITCH

Circular Pitch	Diametral Pitch	Circular Pitch	Diametral Pitch
2 inches.	1.571	7/8 inch.	3.590
1 7/8 "	1.676	13/16 "	3.867
1 3/4 "	1.795	3/4 "	4.189
1 5/8 "	1.933	11/16 "	4.570
1 1/2 "	2.094	5/8 "	5.027
1 7/16 "	2.185	9/16 "	5.585
1 3/8 "	2.285	1/2 "	6.283
1 5/16 "	2.394	7/16 "	7.181
1 1/4 "	2.513	3/8 "	8.378
1 3/16 "	2.646	5/16 "	10.053
1 1/8 "	2.793	1/4 "	12.566
1 1/16 "	2.957	3/16 "	16.755
1 "	3.142	1/8 "	25.133
15/16 "	3.351	1/16 "	50.266

teeth in each 3.1416 inches of circumference, then the circular pitch must equal 3.1416 divided by the diametral pitch, which proves formula (1).

It may not be actually necessary to show how we obtain the diametral pitch from the circular pitch, but we will endeavor to explain everything as we go along. As in the preceding case, we begin with the ratio of the circumference of the circle to its diameter, which is 3.1416. In each 3.1416 inches of circumference we have a certain number of teeth, which is the diametral pitch of the gear. Now, having given the circular pitch, if we divide 3.1416 by that, we obtain the number of teeth for 3.1416 inches of the circumference, which is the diametral pitch of the gear, which proves formula (2).

The accompanying tables will facilitate the finding of corresponding diametral and circular pitches. Table III gives the even diametral pitches with the corresponding circular pitches, while Table IV gives the even circular pitches with the corresponding diametral pitches.

Pitch Diameter

Having given the diametral pitch and number of teeth in a gear, to find the pitch diameter, divide the number of teeth by the pitch, and the quotient will be the pitch diameter, which, expressed in its simplest form, is:

$$\frac{N}{P}=D \qquad (3)$$

in which N = number of teeth; P = pitch (diametral); D = pitch diameter.

Example.—A 10-pitch gear has 35 teeth, what is the pitch diameter? Divide 35 (the number of teeth) by 10 (the pitch), and the quotient $3\frac{1}{2}$ is the pitch diameter of the gear.

The definition of diametral pitch proves this formula. If the diametral pitch equals the number of teeth to each inch of pitch diameter, then dividing the number of teeth in the gear by the diametral pitch will give the number of inches of the pitch diameter. If the circular pitch and number of teeth are given, first find the diametral pitch, and proceed as given above.

Addendum

The addendum of a gear tooth is the distance from the pitch circle to the outside circumference of the gear. This distance is always equal to the reciprocal of the diametral pitch, or 1 divided by the diametral pitch, and, expressed as a formula, is:

$$S=\frac{1}{P} \qquad (4)$$

in which S = addendum; P = diametral pitch.

Outside Diameter

When we start to make a gear, we first wish to know the outside diameter. If we have the pitch and number of teeth given, this may easily be found by the following rule: Add 2 to the number of teeth,

and divide by the pitch. This, expressed as a formula, is:

$$\frac{N+2}{P}=O \qquad (5)$$

in which N = number of teeth; P = diametral pitch; O = outside diameter.

Example.—Given a gear of 20 teeth and 4 pitch, to find the outside diameter. The number of teeth, 20, plus 2 equals 22, and 22 divided by 4 (the pitch of the gear) equals 5½, the outside diameter of the gear.

This formula is simply a combination of formulas (3) and (4), for we first find the pitch diameter, and then add the addendum twice, for it must be added on each side of the pitch diameter. The mathematical solution is as follows:

$$\frac{N}{P}=D;\ D+\frac{1}{P}+\frac{1}{P}=O$$

$$O=D+\frac{2}{P};\ O=\frac{N+2}{P} \qquad (5)$$

Dedendum and Clearance

The dedendum is the working depth of the tooth below the pitch line, and must be equal to the addendum or $\frac{1}{P}$, for the pitch circles of two gears are tangent (touching), so the addendum of one will give the working depth of the other below the pitch line. The clearance is the distance from the end of the dedendum to the bottom of the space between the teeth. There is no common standard for this distance, different gear makers using different distances, yet the difference between them is very slight.

The Brown & Sharpe formula for this distance is:

$$F=\frac{0.157}{P} \qquad (6)$$

in which F = clearance; P = diametral pitch.

The Geo. B. Grant formula is:

$$F=\frac{S}{8} \qquad (7)$$

in which F = clearance; S = addendum.

Thickness of Tooth

The thickness of tooth and width of the space of a gear are always equal at the pitch line, and if the circular pitch is the distance from the center of one tooth to the center of the next tooth measured on the pitch line, tooth and space being equal, then the thickness of tooth must be equal to one-half the circular pitch, or

$$T=\frac{P'}{2} \qquad (8)$$

in which T = thickness of tooth at pitch line; P' = circular pitch.

We know by formula (1) that

$$P' = \frac{3.1416}{P} \qquad (1)$$

and substituting this value for P' in formula (8) we have:

$$T = \frac{\frac{3.1416}{P}}{2}$$

and this formula resolved to its simplest form is:

$$T = \frac{1.5708}{P} \qquad (9)$$

in which T = thickness of tooth at pitch line; P = diametral pitch.

Example.—Given a gear 1 3/16 circular pitch, what is the thickness of tooth at the pitch line? 1 3/16 (the circular pitch) divided by 2 gives 19/32, the thickness of tooth at the pitch line.

Example.—Given a 6-pitch gear to find the thickness of tooth at the pitch line. 1.5708 divided by 6 (the diametral pitch of the gear) gives 0.262, the thickness of tooth at the pitch line.

Table V gives the thickness of tooth at the pitch line for the different diametral pitches.

Depth of Tooth

After we get the gear blank turned up, we next want to know how deep to run the gear cutter in order to get a perfect tooth. The working depth of the tooth we have shown to be equal to the sum of the addendum and dedendum, or $\frac{1}{P} + \frac{1}{P} = \frac{2}{P}$, and the whole depth of the tooth must equal $\frac{2}{P}$ plus the clearance.

Using the Brown & Sharpe standard, we have $\frac{2}{P} + \frac{0.157}{P} =$

$$W = \frac{2.157}{P} \qquad (10)$$

in which 1 = whole depth of tooth; P = diametral pitch.

Example.—Given a gear of 6 diametral pitch, to find the depth of cut to be taken to get a perfect gear tooth.

Divide 2.157 by 6 (diametral pitch) and the quotient 0.359 is the depth to be cut in the gear.

If we had the circular pitch given, to find the depth of tooth, we could substitute in formula (10) the value of P as given in the formula (2), and we would have

$$W = \frac{2.157}{3.1416 \div P'}$$

which, reduced to its simplest form, is:

$$W = 0.6866\, P' \qquad (11)$$

in which W = depth to be cut in gear;

P' = circular pitch.

Example.—Given a gear 1½ inch circular pitch, to find the depth to be cut.

Multiply 0.6866 by 1½ (circular pitch), and the product 1.030 is the depth to be cut in gear.

Table VI gives the depth to be cut in a gear for different diametral pitches.

TABLE V. THICKNESS OF TOOTH AT PITCH LINE

Diametral Pitch	Thickness of Tooth at Pitch Line	Diametral Pitch	Thickness of Tooth at Pitch Line
2	0.785 inch.	12	0.131 inch.
2¼	0.697 "	14	0.112 "
2½	0.628 "	16	0.098 "
2¾	0.570 "	18	0.087 "
3	0.523 "	20	0.079 "
3½	0.448 "	22	0.071 "
4	0.393 "	24	0.065 "
5	0.314 "	26	0.060 "
6	0.262 "	28	0.056 "
7	0.224 "	30	0.052 "
8	0.196 "	32	0.049 "
9	0.175 "	36	0.044 "
10	0.157 "	40	0.039 "
11	0.143 "	48	0.033 "

Distance Between Centers

Having given the number of teeth and diametral pitch of two gears, to find the distance between centers, add the number of teeth together, and divide by twice the diametral pitch, or

$$\frac{N_g + N_p}{2P} = C \qquad (12)$$

in which N_p = number of teeth in one gear,

N_g = number of teeth in other gear,

P = diametral pitch,

C = distance between centers.

This formula is obtained from formula (3):

$$\frac{N}{P} = D$$

This formula gives us the pitch diameter of one gear, and, if we get the pitch diameters of two gears and add them together, we have twice the distance between centers, for the sum of the pitch diameters is twice the sum of the pitch radii, which is the distance between centers.

We have now traced, by the aid of a few "rules," the proportions of a gear tooth, having given the pitch and number of teeth, through

pitch diameter, addendum, dedendum, clearance, width of tooth and depth to be cut, up to the distance between centers. We now give some formulas for the solution of problems in which some of the quantities which were known in preceding problems are unknown.

Pitch

1. To find the pitch, having given the pitch diameter and number of teeth. Divide the number of teeth by the pitch diameter, and the

TABLE VI. DEPTH OF TOOTH

Diametral Pitch	Depth to be cut in gear	Diametral Pitch	Depth to be cut in gear
2	1.078 inch.	12	0.180 inch.
2¼	0.958 "	14	0.154 "
2½	0.863 "	16	0.135 "
2¾	0.784 "	18	0.120 "
3	0.719 "	20	0.108 "
3½	0.616 "	22	0.098 "
4	0.539 "	24	0.090 "
5	0.431 "	26	0.083 "
6	0.359 "	28	0.077 "
7	0.308 "	30	0.072 "
8	0.270 "	32	0.067 "
9	0.240 "	36	0.060 "
10	0.216 "	40	0.054 "
11	0.196 "	48	0.045 "

quotient will be the pitch. The proof of this assertion is derived from the formula:

$$D = \frac{N}{P} \tag{3}$$

If the pitch diameter equals the number of teeth divided by the pitch, then the pitch diameter multiplied by the pitch must equal the number of teeth; therefore the pitch must equal the number of teeth divided by the pitch diameter, and this, expressed in its simplest form, is:

$$P = \frac{N}{D} \tag{13}$$

in which $P =$ pitch (diametral); $N =$ number of teeth in gear; $D =$ pitch diameter.

Example.—A gear, 3 inches pitch diameter, has 36 teeth. Find the diametral pitch.

Divide 36 (the number of teeth) by 3 (the pitch diameter), and we have 12, the diametral pitch of the gear.

2. Having given the outside diameter and number of teeth, to find the diametral pitch. Add 2 to the number of teeth, and divide by the outside diameter, and the quotient will be the pitch of the gear.

In formula (5) we have:

$$\frac{N+2}{P} = O \tag{5}$$

If the number of teeth + 2 divided by the pitch equals the outside diameter, then the outside diameter multiplied by the pitch must equal the number of teeth + 2, and then the pitch must equal the number of teeth + 2 divided by the outside diameter, which, expressed as a formula, is:

$$\frac{N+2}{O}=P \qquad (14)$$

in which N = number of teeth in gear; O = outside diameter; P = diametral pitch.

Example.—Given a gear of 36 teeth and 3 1/6-inch outside diameter; to find the diametral pitch.

36 (the number of teeth) + 2 = 38.

38 ÷ 3 1/6 = 12, the diametral pitch of the gear.

Pitch Diameter

1. Having given the outside diameter and the pitch, to find the pitch diameter. The distance from the pitch diameter to the outside diameter is $\frac{1}{P}$, as explained in formula

$$S=\frac{1}{P} \qquad (4)$$

and as this is to be added on each side of the center, the outside diameter of the gear must be equal to the pitch diameter plus $\frac{2}{P}$. If this is so, then $\frac{2}{P}$ subtracted from the outside diameter will give the pitch diameter, or

$$D=O-\frac{2}{P} \qquad (15)$$

in which D = pitch diameter; O = outside diameter; P = diametral pitch.

Example.—Given a gear 3 1/6 inches outside diameter and 12 pitch; To find the pitch diameter.

3 1/6 (the outside diameter) — 2/12 = 3 inches, the pitch diameter of the gear.

2. Having given the outside diameter and number of teeth, to find the pitch diameter. Multiply the outside diameter by the number of teeth, and divide by the number of teeth plus 2.

We have shown in formula (5) that the outside diameter equals the number of teeth + 2 divided by pitch, or

$$O=\frac{N+2}{P} \qquad (5)$$

and in the formula (13) that pitch equals the number of teeth divided by the pitch diameter, or

$$P = \frac{N}{D} \qquad (13)$$

Now, if the outside diameter equals the number of teeth plus 2 divided by the diametral pitch (and the diametral pitch equals the number of teeth divided by the pitch diameter), then the outside diameter must be equal to the number of teeth plus 2, divided by a fraction with the number of teeth as numerator and the pitch diameter as denominator. This is simply substituting the value of the pitch as shown in formula (13) for the pitch in formula (5), and expressed as a formula, is:

$$O = \frac{N + 2}{N \div D}$$

Multiplying both sides of the equal sign by $\frac{N}{D}$ we have

$$O \times \frac{N}{D} = N + 2, \text{ or } \frac{O \times N}{D} = N + 2,$$

and now, multiplying both sides by D, we have

$$O \times N = (N + 2) \times D$$

and dividing both sides by $N + 2$ we get

$$\frac{O \times N}{N + 2} = D, \text{ or } D = \frac{O \times N}{N + 2} \qquad (16)$$

in which D = pitch diameter; N = number of teeth; O = outside diameter.

Example.—Given a gear 3 1/6 inches outside diameter and 36 teeth. to find the pitch diameter.

3 1/6 (the outside diameter) multiplied by 36 (the number of teeth) equals 114. 36 (the number of teeth) + 2 = 38. 114 ($O \times N$) divided by 38 ($N + 2$) = 3 inches, the pitch diameter of the gear.

Number of Teeth

1. Having given the pitch diameter and pitch, to find the number of teeth. Multiply the pitch diameter by the pitch, and the product will be the number of teeth in the gear.

The diametral pitch of a gear equals the number of teeth for each inch of pitch diameter; hence, if we multiply the pitch by the number of inches of pitch diameter we will have the number of teeth in the gear, which, expressed as a formula, is:

$$N = P \times D \qquad (17)$$

in which P = diametral pitch; D = pitch diameter.

Example.—Given a gear 3 inches pitch diameter and 12 diametral pitch, to find the number of teeth. 3 (pitch diameter) multiplied by 12 (diametral pitch) = 36, the number of teeth in the gear.

2. To find the number of teeth, having given the outside diameter and pitch. Multiply the outside diameter by the pitch and subtract 2, or

$$N = (O \times P) - 2 \qquad (18)$$

CHART FOR DIMENSIONS OF SPUR GEARS

No.	To Find	Rule	Formula
1	Diametral Pitch	Divide 3.1416 by circular pitch	$P = \frac{3.1416}{P'}$
2	Circular Pitch	Divide 3.1416 by diametral pitch	$P' = \frac{3.1416}{P}$
3	Pitch Diameter	Divide number of teeth by diametral pitch	$D = \frac{N}{P}$
4	Pitch Diameter	Multiply number of teeth by circular pitch and divide the product by 3.1416	$D = \frac{NP'}{3.1416}$
5	Center Distance	Add the number of teeth in both gears and divide the sum by two times the diametral pitch	$C = \frac{N_g + N_p}{2P}$
6	Center Distance	Multiply the sum of the number of teeth in both gears by circular pitch and divide the product by 6.2832	$C = \frac{(N_g + N_p)P'}{6.2832}$
7	Addendum	Divide 1 by diametral pitch	$S = \frac{1}{P}$
8	Addendum	Divide circular pitch by 3.1416	$S = \frac{P'}{3.1416}$
9	Clearance	Divide 0.157 by diametral pitch	$F = \frac{0.157}{P}$
10	Clearance.	Divide circular pitch by 20	$F = \frac{P'}{20}$
11	Whole Depth of Tooth	Divide 2.157 by diametral pitch	$W = \frac{2.157}{P}$
12	Whole Depth of Tooth	Multiply 0.6866 by circular pitch	$W = 0.6866 P'$
13	Thickness of Tooth	Divide 1.5708 by diametral pitch	$T = \frac{1.5708}{P}$
14	Thickness of Tooth	Divide circular pitch by 2	$T = \frac{P'}{2}$
15	Outside Diameter	Add 2 to the number of teeth and divide the sum by diametral pitch	$O = \frac{N+2}{P}$
16	Outside Diameter	Multiply the sum of the number of teeth plus 2 by circular pitch and divide the product by 3.1416	$O = \frac{(N+2)P'}{3.1416}$
17	Diametral Pitch	Divide number of teeth by pitch diameter	$P = \frac{N}{D}$
18	Circular Pitch	Multiply pitch diameter by 3.1416 and divide by number of teeth	$P = \frac{3.1416\,D}{N}$
19	Pitch Diameter	Subtract two times the addendum from outside diameter	$D = O - 2S$
20	Number of Teeth	Multiply pitch diameter by diametral pitch	$N = P \times D$
21	Number of Teeth	Multiply pitch diameter by 3.1416 and divide the product by circular pitch	$N = \frac{3.1416\,D}{P'}$
22	Outside Diameter	Add two times the addendum to the pitch diameter	$O = D + 2S$
23	Length of Rack	Multiply number of teeth in rack by 3.1416 and divide by diametral pitch	$L = \frac{3.1416\,N}{P}$
24	Length of Rack	Multiply the number of teeth in the rock by circular pitch	$L = NP'$

in which N = number of teeth; O = outside diameter; P = diametral pitch.

This formula is simply the reverse of formula

$$\frac{N+2}{P}=O \qquad (5)$$

If the outside diameter equals the number of teeth + 2 divided by the pitch, which we have already proved, than the number of teeth plus 2 must equal the outside diameter multiplied by the pitch, and subtracting 2 from this result we have the number of teeth in the gear.

Example.—Given a gear 3 1/6 inches outside diameter and 12 pitch, to find the number of teeth. Multiply 3 1/6 (outside diameter) by 12 (the pitch) and we have 38, and subtracting 2 from this result we have 36, the number of teeth in the gear.

Outside Diameter

To find the outside diameter having given the pitch diameter and pitch. Divide 2 by the pitch and add to the pitch diameter, or

$$O=D+\frac{2}{P} \qquad (19)$$

in which O = outside diameter,
D = pitch diameter,
P = pitch.

The addendum of a gear is $\frac{1}{P}$ [formula (4)] and this, added on each side of the pitch diameter, gives the outside diameter.

Example.—Given a gear 3 inches pitch diameter and 12 pitch; to find the outside diameter.

3 (pitch diameter) plus 2/12 $\left(\frac{2}{P}\right)$ = 3 1/6 inches, the outside diameter of the gear.

Summary of Formulas

In the chart on page 20, the rules and formulas for the dimensions of spur gears have been grouped together, so that they may be more easily found when wanted. The same reference letters are used as have previously been employed in deducing the various formulas. It will be noticed that with the aid of the formulas given in the chart, each dimension can be calculated either from the diametral or circular pitch. The first sixteen formulas are placed in the order in which they would naturally present themselves to the designer when determining the dimensions of a pair of spur gears. Nos. 17 to 22 give additional formulas for various conditions of known and unknown factors. Formulas Nos. 23 and 24 are for racks.

CHAPTER III

INTERNAL SPUR GEARS

As indicated by its name, the internal gear is one having teeth formed on an interior pitch surface instead of an exterior one. In a word, it is an ordinary spur gear turned inside out. At the right of Fig. 10 is shown a sketch of such a gear, meshing with a spur pinion; at the left is shown a pair of spur gears having the same number and pitch of teeth as the pinion and internal gear on the right. By tracing the motion in each figure, it will be seen that internal action causes the

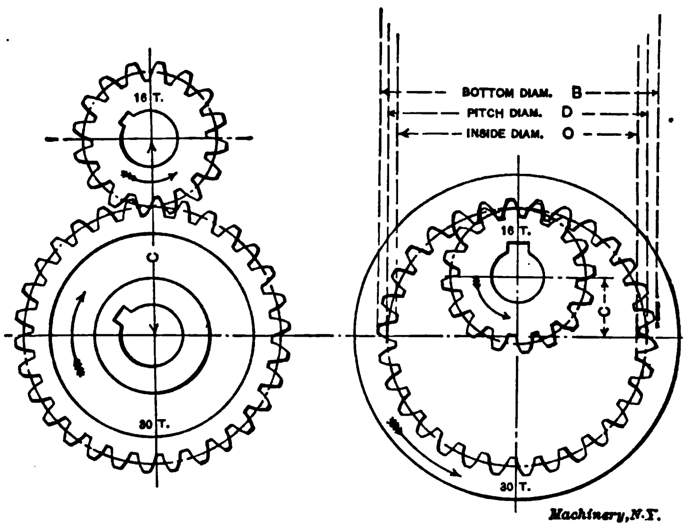


Fig. 10. Internal Spur Gears

two members to turn in the same direction, while external action produces opposite rotation.

The Uses of Internal Gearing

There are some advantages attaching to the use of internal gears for particular applications, as compared with external gears of the same pitch and number of teeth. For one thing, an internal gear has its teeth and that of its pinion protected to a very marked degree from inflicting or receiving injury, often making the use of a gear guard unnecessary if the parts are properly designed for that purpose. Owing to the fact that the cylindrical pitch surfaces in internal gearing have their curvature in the same direction, the teeth of the pinion approach and mesh with those of its mate somewhat more gradually and easily than when they are meshing with an external gear. This tends toward smoothness and quietness in running, as well as giving a slight-

ly longer contact for each tooth. Another characteristic which is often an advantage will be seen from a study of Fig. 10. In each case shown we have gears of the same pitch and number of teeth. The internal gears evidently figure out to a much smaller center distance than the external gears. This matter is of importance when it is necessary to transmit considerable power between shafts placed quite close together.

In contrast with the advantages just mentioned, the chief factor which has limited the use of the internal gear, has been the difficulty and expense of making it. This difficulty has not been insuperable for cast gearing, but, until the introduction of recent processes, the cutting of internal teeth has been tedious and unsatisfactory.

Rules for Designing Internal Gearing

Neglecting for the time being the modifications which have to be made in the standard proportions to get rid of interference, it may be said that the usual thing to do in designing internal gearing is to follow exactly the dimensions of the standard system as used for external spur gearing. Practically the only modifications required in the rules given on page 20 are those made necessary by the fact that the center distance, in internal gearing, is equal to the difference between the two pitch radii, instead of to their sum. Besides this, we have of course to reckon with the fact that the teeth are turned inside out, so that the bottom or root diameter is larger than the pitch diameter.

The only new term is "Inside Diameter," which takes the place of the outside diameter of external spur gearing. It is, of course, the inside diameter of the blank before the teeth are cut, and it is marked *O* in Fig. 10. The following are the rules which must be changed:

No. 5 will read: *To find the center distance, subtract the number of teeth in the pinion from the number of teeth in the gear and divide the remainder by* 2 *times the diametral pitch.*

No. 6 will read: *To find the center distance, multiply the difference of the numbers of teeth in the gear and pinion by the circular pitch and divide the product by* 6.2832.

No. 15 will read: *To find the inside diameter, subtract* 2 *from the number of teeth and divide the remainder by the diametral pitch.*

No. 16 will read: *To find the inside diameter, subtract* 2 *from the number of teeth, multiply the remainder by the circular pitch, and divide the product by* 3.1416.

No. 19 will read: *To find the pitch diameter, add twice the addendum to the inside diameter.*

No. 22 will read: *To find the inside diameter, subtract twice the addendum from the pitch diameter.*

Interference

In laying out the shape of teeth for internal gearing we have to look out for two kinds of interference which are almost sure to be met with. The points of the rack teeth in the 14½-degree involute system are relieved to avoid the interference with the flanks of small pinions, and the points of internal gear teeth have to be relieved for the same

reason, but to an even greater degree. A second form of interference occurs when the pinion has too nearly the same number of teeth as the gear. In this case there is a tendency for the points of the pinion and the gear teeth to strike as they roll into and out of engagement.

It would require an extended treatise to explain the different ways of avoiding these two forms of interference. For the first form correction may be made either by correcting the points of the internal gear tooth by shortening them (at the same time preferably lengthening the addendum of the pinion), or by changing the pressure angle. The mechanic who desires to use internal cut gearing without making a study of the theoretical conditions has two courses open to him. He may purchase a formed cutter for the gear from the regular makers of formed cutters, telling them the number of teeth and pitch he proposes to use for the gear and pinion. In that case, the maker of the cutter will make such corrections in its form as may be necessary to avoid interference. Another way is to cut the internal gear on the Fellows gear shaper. With this machine the cutter forms its own correction, so that no calculation on the part of the user is ordinarily required.

CHAPTER IV

STRENGTH AND DURABILITY OF SPUR GEARS

The principal materials used for gears are cast iron, steel (both from bar stock and in the form of forgings or castings), brass and bronze (from stock or castings), rawhide, and fiber.

Cast iron is, perhaps, the most used material. It is one of the cheapest, and is the easiest to mold or cut to shape. It wears fairly well also. Its greatest disadvantages are its lack of resisting power to shock or impact, and the uncertainty of the quality of the casting into which it is formed. A casting is liable to various more or less serious defects, some of which may be visible from the exterior, while others are concealed; 'blow holes," "cold shuts," "scabs," etc.," are of common occurrence where the foundry work is not skillfully done. Castings from iron are so cheap, however, that those containing such defects may be discarded without hesitation as soon as they are discovered. The prime advantage of the material, aside from its cheapness, is the facility with which it may be molded into any desired form, so that we may have large gears with arms, webs, projecting bosses, counterbalances, etc., to suit the mechanism being designed. These advantages will doubtless continue to keep cast iron the most used of all materials for ordinary work.

In a pair of gears the pinion is often made of steel, from bar stock or a drop forging, even when the larger gear is made from cast iron or some other material. Steel has the advantage over cast iron of being more resilient—that is, it offers a greater resistance to shock

or impact. Since the pinion of a pair of gears naturally has teeth of a weaker form than those of its mate, it should be made of stronger material. Furthermore, there is usually less friction between two different metals in contact than between two parts made of the same material. Still further, the smaller of a pair of gears will wear out much faster than its mate, as each of its teeth is in action a greater number of times in a given period; so on this account as well, it should be made of the more resisting material. As the softer grades of steel, however, are not very durable, steel pinions are sometimes case-hardened, or they may be made of high carbon steel that can be hardened without requiring the action of carbonizing materials. By such processes the pinions gain in durability, but suffer somewhat in accuracy of outline, since the natural result of heat treatment of any kind is to warp and distort the part treated. It is possible to grind hardened gears to the correct outline, leaving a little stock on the sides of the teeth, after cutting, for this purpose. There would, however, seem to be some doubt of the commercial success of the process, owing to the difficulty of keeping the grinding wheel to its proper shape, and keeping the mechanism of the machine itself in proper condition in the presence of the emery dust with which it is surrounded.

Steel blanks for medium sized gears are sometimes drop-forged to bring the wheel to the desired shape. Fairly long hubs, a thickened rim, and a thin web, can be formed in this way without requiring the form to be turned out of solid metal, with the attending waste of time and material. The steel drop forging, thus, has some of the advantages of the casting. It is more costly than the casting, but may be made of better material.

When gears are made of steel from the bar or from forgings, a wide range of physical qualities is offered. The steel may have almost any desired strength, hardness and resilience, or capacity to resist shock. The matter of hardness is important in the case of high-speed gearing. Two soft materials working on each other at a high velocity tend to abrade each other, so one at least of a pair of gears so used should be of a fairly hard grade of steel.

When it is desired to extend to large gears the advantages resulting from the use of steel, making the blanks from round stock or from forgings becomes impracticable, so the use of steel castings is resorted to. This is the best material for large heavy-duty gears. The art of making castings from steel, clean and sound throughout, has only become understood in the past few years. Blow holes, and rough, dirty casting are still common enough, and the use of this material still labors somewhat under the disadvantage of slow deliveries, due to the fact that the steel foundries are not so widely distributed that each machine builder can find one in his own town, as is usually the case with the iron foundry. What has been said as to the qualifications of bar and forged steel as a material for gears, applies also to steel castings, except that there is a narrower range of choice in qualities of the metal, the higher grades not being available for castings.

In general, it may be said that there is a growing tendency to sub-

stitute steel in some form for cast iron as a material for gears. This tendency is especially marked in machine tool design.

It is common and good practice to use a composition metal like brass or bronze for the smaller of a pair of lightly loaded gears which have to run at high speed. When such gears are run with a large gear of cast iron, the difference in texture between the two materials used lessens the friction, and there is a gain on the score of noiselessness as well. Brass may be used where the duty is very light; higher grades of material, like phosphor bronze, are used for heavier service at high speed. When the service becomes quite severe, the materials in the gears should be reversed, so that the larger one is of phosphor bronze, and the smaller one of steel. The pinion has thus its maximum of strength and durability, at the same time that the advantages resulting from the use of dissimilar materials are retained.

Where noiselessness is a prime consideration, rawhide is extensively used. This non-metallic substance possesses the required structure to deaden the sound vibrations, together with a considerable degree of toughness, when properly cured. Manufacturers of gear blanks from this material cure the hide by processes which they claim give far better results for this service than can be obtained by ordinary means. The material is not injured by oil, though it does not require lubrication in service; but there has been some complaint of its swelling and losing its shape when exposed to moisture. Trouble from this source may, however, have been due to the use of an inferior grade of material, because thousands of rawhide pinions are in satisfactory daily use, under all sorts of conditions, at the present time. It is a somewhat more costly material than the others commonly used, but its compensating freedom from noise is often worth more than the added expense. But one of a pair of gears—generally the pinion—is made from this substance, the gear being of steel or iron. Gears as large as 40 inches in diameter have been made from this material.

Fiber is another material used under about the same conditions as rawhide. It is not as strong, and it suffers under the disadvantage of being difficult to machine, owing to its peculiar gritty structure. It is also liable to swell in the presence of moisture. It has an advantage over rawhide in that it is comparatively inexpensive, and may be purchased in a variety of sizes of bars, rods, tubes, etc., so that it is convenient to use at short notice. For light duty at high speed it serves its purpose very well.

Racks of large size, such as those used for driving the platens of metal planers, are made of iron or steel castings. Smaller ones are made from bar steel stock, either machine steel finished all over, or cold rolled steel. The latter material does not require other machining than the cutting of the teeth, being accurately finished to certain convenient sizes in the process of rolling. The cutting of the teeth causes the stock to bend, however, necessitating a straightening operation.

Strength of Gear Teeth

The rule in most common use for determining the strength of gears is the one proposed by Mr. Wilfred Lewis, and described by him in a

paper read before the Engineers' Club of Philadelphia. The utility of this rule is due to its simple form, to the fact that it takes into account a greater number of factors than does any other, and to the fact that the effect of each of these factors is rationally expressed in the formula.

Table VII may be used for finding the allowable unit fiber stress to use in the Lewis formula, for any given speed, for different materials. The values for steel and cast iron are those originally suggested, altered slightly to agree with formula (2), given in the chart on page 29; those for phosphor bronze have been added by the author. It will be noted that two columns of values are given for each material; as

TABLE VII. WORKING STRESSES FOR USE WITH THE LEWIS FORMULA

		Safe Working Unit Stress = S					
		Cast Iron		Phosphor Bronze		Steel	
Velocity in Feet per Minute = V	Strength Factors	Ordinary Workmanship	High-grade Workmanship	Ordinary Workmanship	High-grade Workmanship	Ordinary Workmanship	High-grade Workmanship
0	1.000	6000	8000	9000	12000	15000	20000
100	0.857	5150	6850	7700	10300	12800	17100
200	0.750	4500	6000	6750	9000	11200	15000
300	0.666	4000	5350	6000	8000	10000	13300
450	0.571	3400	4550	5150	6850	8550	11400
600	0.500	3000	4000	4500	6000	7500	10000
900	0.400	2400	3200	3600	4800	6000	8000
1200	0.333	2000	2650	3000	4000	5000	6650
1800	0.250	1500	2000	2250	3000	3750	5000
2400	0.200	1200	1600	1800	2400	3000	4000

explained, the first set of values may be used for workmanship of ordinary grade, while the other is permissible with a higher grade. The second column, of strength factors, may be used for finding the allowable working fiber stress to use for any given speed, when the safe static stress is known; to find the fiber stress, multiply the safe static stress by the strength factor. Formula (2) on page 29 may be used for the same purpose, giving results closely approximating the values of Table VII.

The variable factor introduced into the problem by the varying shape of teeth of the same pitch in gears of different numbers of teeth, is taken care of by introducing in the formula the outline factors Y given in the chart on page 29. These factors are given for that arrangement of the formula which applies to diametral pitches.

The formulas in the chart take no account of any such limitation of the width of face for a gear of given pitch as obtains in practice. The strength is made to increase directly with the width of face, without limit. In practice, if the face is too long in proportion to the size of the tooth, we cannot be sure that each tooth of the gear has a full

bearing over its whole length on its mating tooth in the other gear. The shafts on which the two are mounted may not be parallel, or, if originally parallel, they may deflect under the strain of the load transmitted. For reasons like this, in ordinary commercial work, the width of face may be considered as well proportioned when it equals, in inches, 8.75 divided by the diametral pitch. As a formula, this gives us $A = \frac{8.75}{P}$.

In cases where accurate workmanship can be depended on, there is a gain in using teeth of wider face and finer pitch than would be allowed by the formulas on page 29. There is a gain in efficiency, smoothness of action, and noiselessness, especially at high speeds. In fact, the width of face of a gear may well be made to depend in part on the speed at which it is run, as well as on the pitch, it being taken for granted, of course, that the pitch and width are such as to give the required strength. A suggested relation between the speed and the face is expressed in the following rule, offered by an English engineer as having given good results in an extended practice. It has been changed to derive the answer directly from the diametral pitch, instead of from the circular pitch, as originally given: *To find a well-proportioned width of face for carefully made gearing, multiply the square root of the pitch line velocity in feet per minute by* 0.15, *add* 9 *to the product, and divide the result by the diametral pitch.* As a formula, this gives us:

$$A = \frac{0.15\sqrt{V} + 9}{P}$$

Example.—What should be the pitch and the width of face of a steel pinion, 4 inches in pitch diameter, the teeth shaped according to the 14½-degree involute system, running 750 revolutions per minute, and transmitting 10 horse-power; the workmanship is high grade, and the width of the face is to be proportioned according to the rule and formula given immediately above?

The velocity at the pitch line is $= 0.262\,D\,R = 0.262 \times 4 \times 750 =$ 786 feet per minute. (See formula (1) in the chart for strength of spur gears.) The allowable running stress for a static stress of 20,000 pounds per square inch is found by formula (2):

$$S = 20{,}000 \times \frac{600}{600 + 786} = 8{,}660 \text{ pounds per square inch.}$$

The load at the pitch line is equal to

$$\frac{10 \times 33{,}000 \times 12}{\pi \times 4 \times 750} = 420 \text{ pounds.}$$

Assume 5 diametral pitch as a trial pitch for the teeth; then the number of teeth equals $5 \times 4 = 20$. Transposing formula (3):

$$W = \frac{S\,A\,Y}{P} \text{ gives us } A = \frac{W\,P}{S\,Y}.$$

List of Reference Letters.

D = pitch diameter of gear in inches.

R = revolutions per minute.

V = velocity in ft. per min. at pitch diameter.

S_S = allowable static unit stress for material.

S = allowable unit stress for material at given velocity.

A = width of face.

Y = outline factor (see table below).

P = diametral pitch (if circular pitch is given, divide 3.1416 by circular pitch to obtain diametral pitch).

C = pitch cone radius.

W = maximum safe tangential load in pounds at pitch diameter.

H.P. = maximum safe horse-power.

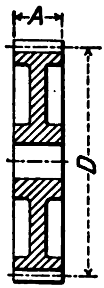


Table of Outline Factors (Y).

Number of Teeth	Outline Factor = Y: 14½° Involute (Std) and Cycloidal	Outline Factor = Y: 20° Involute	Number of Teeth	Outline Factor = Y: 14½° Involute (Std) and Cycloidal	Outline Factor = Y: 20° Involute
12	0.210	0.245	27	0.314	0.349
13	0.220	0.261	30	0.320	0.358
14	0.226	0.276	34	0.327	0.371
15	0.236	0.289	38	0.336	0.383
16	0.242	0.295	43	0.346	0.396
17	0.251	0.302	50	0.352	0.408
18	0.261	0.308	60	0.358	0.421
19	0.273	0.314	75	0.364	0.434
20	0.283	0.320	100	0.371	0.446
21	0.289	0.327	150	0.377	0.459
23	0.295	0.333	300	0.383	0.471
25	0.305	0.339	Rack	0.390	0.484

Use rules and formulas 1 to 4 in the order given.

No.	To Find	Rule	Formula
1	Velocity in ft. per min. at the pitch diameter	Multiply the product of the diameter in inches and the number of revolutions per minute, by 0.262	$V = 0.262DR$
2	Allowable unit stress at given velocity	Multiply the allowable static stress by 600 and divide the result by the velocity in feet per minute plus 600	$S = S_S \times \frac{600}{600+V}$
3	Maximum safe tangential load at pitch diameter	Multiply together the allowable stress for the given velocity, the width of face, the tooth outline factor; divide the result by the diametral pitch	$W = \frac{SAY}{P}$
4	Maximum safe Horse-Power	Multiply the safe load at the pitch line by the velocity in feet per minute, and divide the result by 33,000	$HP = \frac{WV}{33{,}000}$

Apply this transposed formula and find a trial width of face (factor Y is given in the table in the chart):

$$A = \frac{420 \times 5}{8,660 \times 0.283} = 0.9 \text{ inch, approximately.}$$

For 5 pitch, however, according to the formula

$$A = \frac{0.15\sqrt{V} + 9}{P}$$

the width of face should be

$$A = \frac{0.15\sqrt{786} + 9}{5} = 2.64 \text{ inch.}$$

Thus, our pitch is evidently too coarse. Repeated trials show that if we make calculations for 9 diametral pitch the results from the two formulas for width of face agree fairly well. Thus:

$$\text{Number of teeth} = 9 \times 4 = 36.$$

$$A = \frac{420 \times 9}{8,660 \times 0.332} = 1.32 \text{ inch.}$$

$$A = \frac{0.15\sqrt{786} + 9}{9} = 1.4 \text{ inch.}$$

We may, therefore, settle on 9 diametral pitch and 1⅜-inch width of face, as the dimensions to which the gear ought to be made.

It will be noted that nothing was said about rawhide in the table on page 27, giving the allowable stresses for different materials at different velocities. There is not as much information available as might be desired on the strength of rawhide pinions. One prominent firm, the New Process Raw Hide Co., makes a regular practice of replacing high speed cast iron pinions with those made of rawhide, of the same dimensions. Where, however, a rawhide pinion is to replace a steel gear, working under severe conditions, a special construction is used, in which the weaker material is strengthened by a bronze reinforcement.

Mr. Diefendorf, the chief engineer of the New Process Raw Hide Co., gives the following information on this subject: "Our published statement, that cast iron pinions can be replaced with rawhide ones of the same pitch and number of teeth, holds good down to a peripheral velocity of about 1,600 feet per minute. Figuring on a peripheral velocity of 1,800 feet per minute for a 15-tooth pinion, new process rawhide pinions are good for a load of 150 pounds per inch of face for a 1-inch circular pitch gear, with other pitches in proportion, up to a maximum load not to exceed 250 pounds per inch of face, beyond which we have found it undesirable to go, owing to the compression of rawhide under heavy loads. It would appear that the elastic limit and compression point should be taken into consideration more than the tensile strength, as the material will bend long before it shows any sign of breakage; in fact, it is owing to its elastic qualities that

our rawhide is able to withstand shocks at high speed that would possibly strip the teeth of cast iron pinions."

The standard German engineers' handbook, "Hütte," gives a rule which may be translated into the following form for English measurements: *To find the allowable load in pounds at the pitch line for a rawhide pinion, multiply the width of face in inches by from* 180 *to* 360, *and divide the product by the diametral pitch.* It will be seen that this gives much lower permissible loads than does the New Process Rawhide Co.'s rule, which reduces to a factor of about 470, in place of the 180 to 360 given in "Hütte." In both of these rules the strength is made independent of the velocity at the pitch line. Since decrease of strength with increase of velocity is due to impact, and since rawhide is a substance peculiarly fitted to absorb impact harmlessly, it is logical to assume that the effect of increasing the velocity is negligible. This accounts for the fact that a rawhide gear will be as strong as a cast iron one at high speeds, when it would appear very weak in comparison with it in a static test.

Durability of Gearing

A pair of gears figured by the rules we have just given, to be strong enough for the service they are to undergo, may or may not be so proportioned as to be commercially durable. By "commercially durable" gears, we mean those which will last well in comparison with the rest of the machine of which they are a part. In some classes of machinery, gears strong enough for their work would certainly be commercially durable. A rack and pinion, for instance, used to raise a sluice gate for a dam, if made strong enough, would evidently wear indefinitely, though they might rust away. It is plain that all gearing designed for occasional or intermittent use, even under heavy loads, is strong enough to wear well if it is strong enough to bear the load placed upon it. With gearing used for the continuous transmission of power, however, we cannot be sure of this. The gearing of a drive connecting a motor with a printing press, for instance, might conceivably be strong enough and yet not wear as long as the rest of the machine.

The pinion will naturally wear faster than its mate, since each of its teeth is in action a greater number of times per minute. To make the life of the two more nearly alike, it is customary to make them of different materials, as already mentioned, the pinion being made of the more durable one. Thus, a combination of steel pinion and cast iron gear is common and occasionally conditions are found which warrant the expense of a hardened steel pinion and a phosphor bronze gear. The use of the better material in the smaller gear of the pair is proper from the standpoint of strength as well as from that of durability. An examination of the Lewis outline constants as tabulated in the preceding section of this chapter, will show that the teeth of the pinion are always weaker than those of the gear; so it is necessary, if an excess of strength is to be avoided in the gear, to make the pinion of the stronger material; but if the pinion is a little less durable

than the gear, it will take most of the wear; and being more cheaply renewed than its larger mate, the mechanism is kept up at a less expense. It is not wise to use soft steel in both members for heavy service at high speed.

Where the velocity ratio is not extreme and severe service is exacted, as in automobile gearing, the two members may be made of the same material—hardened, or preferably, case-hardened alloy steel.

Efficiency of Standard Spur Gears

The efficiency of two spur gears (or of any other power transmitting mechanism, for that matter) is measured by the percentage they deliver of the power entrusted to them. Thus, if a water wheel delivers 160 horse-power to the driving pinion on the shaft on which it is mounted, and the mating gear on the jack-shaft transmits 140 horsepower to that shaft, the efficiency of the gearing is $140 \div 160 = 87\frac{1}{2}$ per cent.

To obtain the maximum of efficiency, attention must be paid to the following considerations:

Form the teeth as near to the perfect theoretical shape as good workmanship will bring them, giving the acting surfaces a fine smooth finish. If the teeth are milled to shape, special cutters should be used, made accurately to shape for the exact number of teeth. The gears must be mounted firmly and accurately in their working position.

Use hard, close-grained materials, preferably different for the two gears. Hardened steel on phosphor bronze will probably give the best results, though it is difficult to be sure of the exact shape in hardened gears, unless they are finished by grinding after hardening. Soft steel on soft steel is probably the worst combination so far as efficiency is concerned, though it is stronger than cast iron on cast iron.

Provide continuous and copious lubrication, preferably by an oil or grease bath in an enclosed casing. Lubricated gears should always be enclosed if they are exposed to dust or grit in the slightest degree, otherwise they will grind each other away, and might rather run entirely dry.

Use as fine a pitch as possible, without requiring too wide a face to transmit the power required. The smaller the pitch, the greater the efficiency. Anything that tends to shorten the line of contact, confining it to the vicinity of the pitch point, increases the efficiency, as there is more rubbing at the beginning and end of contact than when the teeth are passing the pitch point.

In general, it may be said that there is no method of transmitting power between two parallel shafts that is more efficient than a pair of well designed and constructed spur gears, working under proper conditions. The highest efficiency, of course, is obtainable only at a considerable expense, so judgment is required to know how far it is wise to carry the possible refinements.

CHAPTER V

DESIGN OF SPUR GEARING

Typical designs of wheels for different materials and uses are shown in Figs. 11 to 16. The one shown in Fig. 11 is the usual one for a steel, cast iron, brass, bronze or fiber pinion. All of its proportions are determined by the gear calculations and the diameter of the shaft on which it is mounted, so there is little to be said about the design. When made from steel, it is generally formed from bar stock in the lathe or screw machine; for other metals, cast blanks are mostly used.

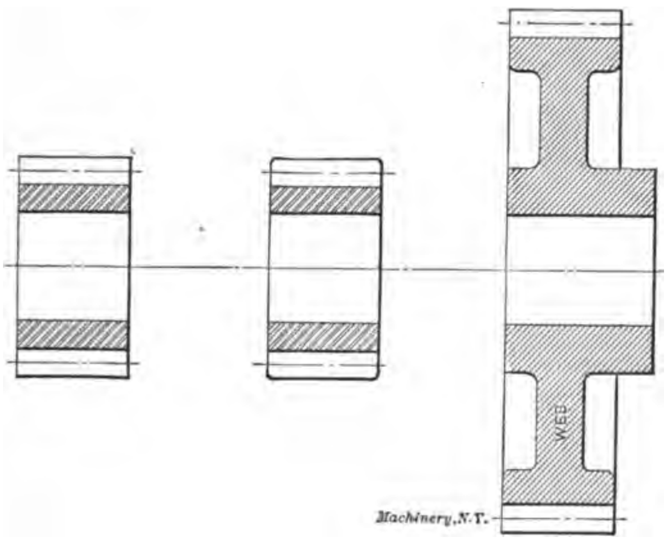
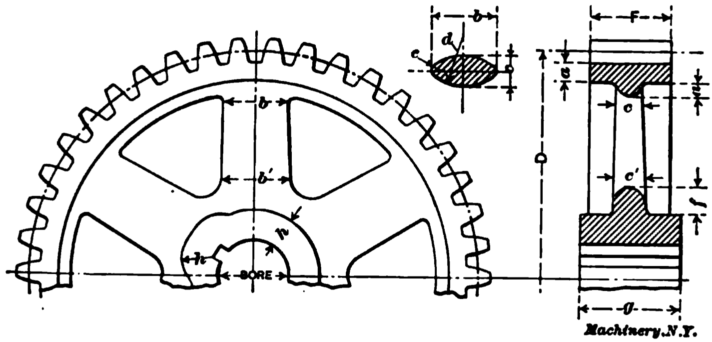


Fig. 11 Fig. 12 Fig. 13

It is the practice of some firms, notably the Brown & Sharpe Mfg. Co. of Providence, R. I., to round the corners of all pinion and gear blanks, large and small, as shown in Fig. 12. This practice has the advantage of making a gear more easy to handle, and less liable to injury in case it is accidentally dropped; it gives it a neater appearance as well.

In Fig. 13 is shown a design used for gears having rather more teeth than those ordinarily known as "pinions." The weight has been lightened by recessing the sides to form the web shown, connecting the rim and the hub. Wheels of this shape are rarely cut from bar stock as the removal of the metal to form the web is too wasteful. The usual practice for this design is to make the blank from a casting or a drop forging.

As the number of teeth for the gear becomes still larger, the increasing weight of the wheel may be lightened by cutting out relieving spaces in the web, or by abandoning the web entirely, and using arms for supporting the rim. This scheme, shown in Fig. 14, with arms of oval section, is the one best adapted for small and medium sized gear



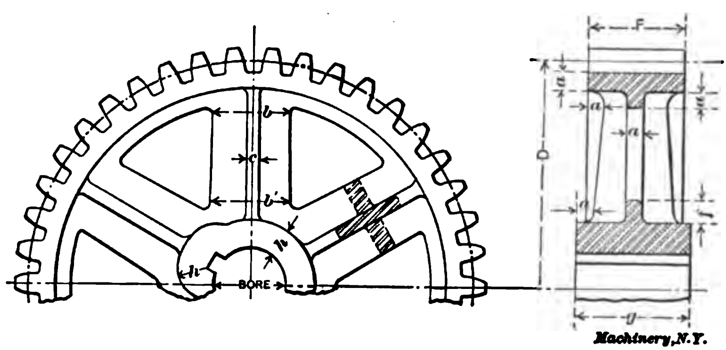
P = diametral pitch, P' = circular pitch.

$a = 1.57 \div P = 0.5\ P'$
$b = 6.28 \div P = 2.0\ P'$
$c = 3.14 \div P = P'$
$d = 4.71 \div P = 1.5\ P'$
$e = 0.79 \div P = 0.25\ P'$

$f = 2.00\ P = 0.65\ P'$
$g = F + 0.025\ D$
$h = 0.44 \times$ bore
$b' = b + \frac{3}{4}$ inch per foot
$c' = c + \frac{3}{4}$ inch per foot

Fig. 14. Dimensions of Spur Gears with Oval Arms

blanks, and is often used on the largest work as well. It is the handsomest of all designs of gear wheels, when it is in harmony with the rest of the machine to which it belongs. It requires somewhat more metal for the same strength than do the two designs next shown. It is



P = diametral pitch, P' = circular pitch.

$a = 1.57 \div P = 0.5\ P'$
$b = 7.85 \div P = 2.5\ P'$
$c = 0.94 \div P = 0.3\ P'$
$f = 2.20 \div P = 0.7\ P'$

$g = F + 0.025\ D$
$h = 0.44 \times$ bore
$b' = b + \frac{3}{4}$ inch per foot

Fig. 15. Dimensions of Spur Gears with Ribbed Arms

very easily molded. Suitable dimensions for wheels of various sizes made in this way, are tabulated below the illustration.

For the largest gears, made of steel, cast iron or bronze castings, wheels with arms of + or H-section are largely used. Dimensions

for wheels of these types are given in Figs. 15 and 16. In these designs, the metal is so distributed as to give a high degree of rigidity for the weight. These two forms, particularly that in Fig. 16, are more difficult to mold than those previously shown. The latter form is better for gears whose faces are very wide in proportion to their pitch, than either of the two in Figs. 14 and 15.

The tabular dimensions given for the various forms of wheels are to be considered as suggestive rather than authoritative. The tables have been in constant use for some years, however, and have proved to be very satisfactory. "Draft," for removing the patterns from the sand in molding, is not shown in any of the illustrations. It should be provided liberally, and should be added to the dimensions given, rather than taken off.

The Governing Conditions in the Design of Gearing

The problem of gear design is one of materials and dimensions. The considerations on which the designer bases his choice of materials and

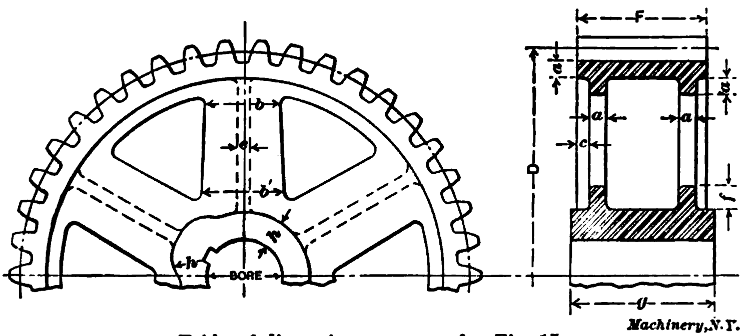


Table of dimensions same as for Fig. 15.

Fig. 16. Spur Gear with Arms of H-section

dimensions are those of strength, durability, efficiency, smoothness of action, noiselessness and cost. The gear cannot attain perfection in all these particulars, as some of them are mutually hostile; the item of cost, especially, has to be sacrificed to make a gain in any other direction. The problem of design is thus one of compromise, and the designer has only his judgment to rely on in determining the relative importance of the various considerations.

It is possible, however, to lay down a few simple rules along this line. The prime consideration is that of strength. If the teeth of the gear are not strong enough to transmit the pressure they are calculated on to bear, the gear will break, and the other virtues it may possess in the way of cheapness, noiselessness, etc., will be of no avail. As has already been stated, in gearing subjected to occasional use only, the durability is sufficient for all practical purposes if the strength is sufficient; but there is a possibility that gearing transmitting power at high speed may wear out before it breaks. Where gearing is used, as in automatic machinery, primarily to obtain certain desired movements in the mechanism, without requiring the transmission of any

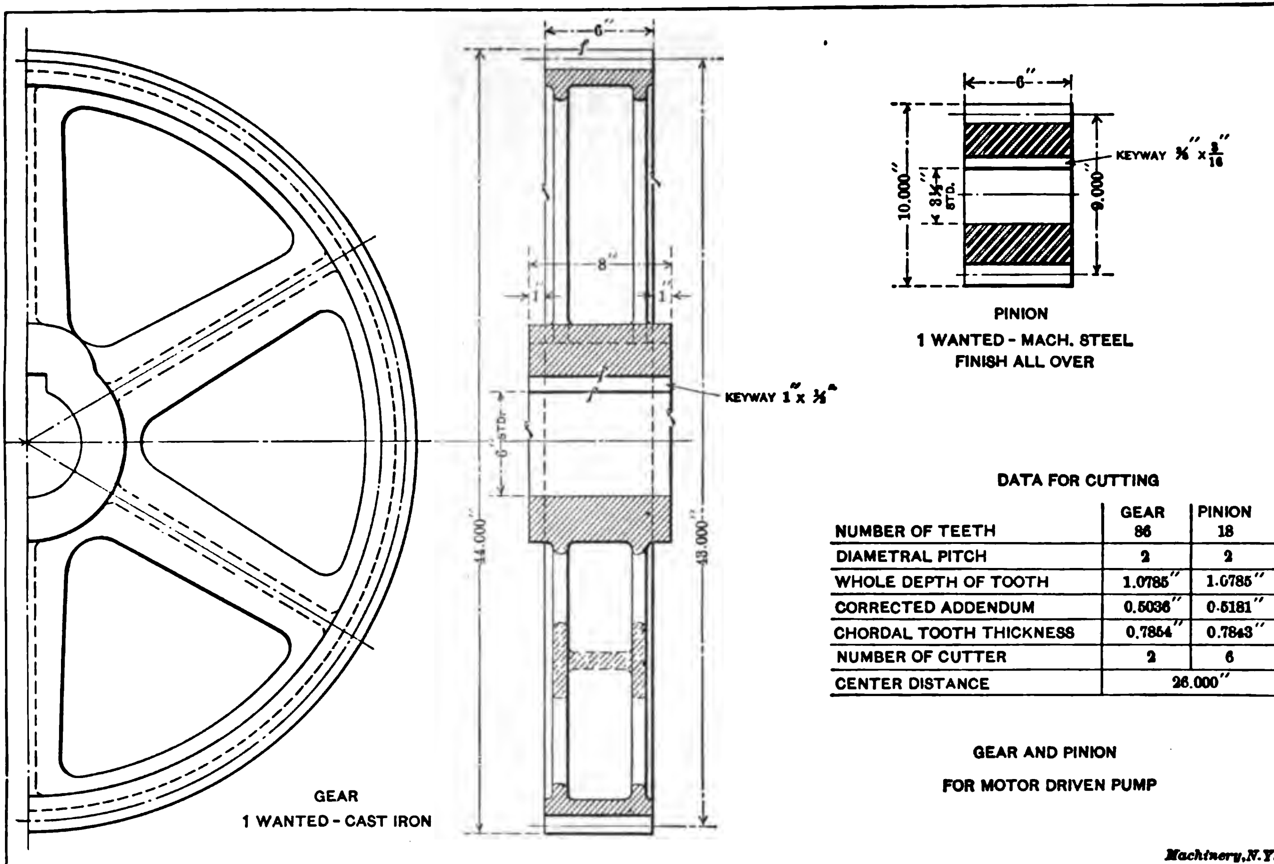
great amount of power, the question of efficiency is not of prime importance. On the other hand, when the main object of a pair of gears is to receive so many horse-power from one shaft, and transmit as nearly as possible the same amount to another shaft, the question of efficiency becomes one worthy of the most careful consideration. As to smoothness of action, if the requirements for efficiency and silence have been met the gears will run smoothly, without shock or vibration. Noiselessness, as a problem in design, is largely a matter of the selection of materials, always supposing that the teeth of the gears are formed to correct tooth curves. The matter of cost is in part an engineering problem, and in part a commercial one. It is an engineering problem in so far as it bears on the problem of obtaining the greatest perfection in the other particulars enumerated, with a given expenditure; and it is a commercial problem when it comes to determining how great a degree of refinement it is advisable to ask the purchaser of the finished machine to pay for.

A Model Spur Gear Drawing

After designing a pair of spur gears with all the care that theoretical knowledge and practical experience can suggest, there still remains the important task of recording the results thus obtained on a drawing, in such a form that they will be intelligible to an intelligent workman. This drawing should plainly set forth every point of information needed for the successful completion of the work. In aiming at this mark, the student should study the model drawing shown in Fig. 17. The arrangement of this drawing, and the amount and kind of information shown on it, are based on the drawing-room practice of the Brown & Sharpe Mfg. Co., Providence, R. I. Some changes and additions, however, have been made by the author. The design of the wheel in Fig. 17 is the same as that shown in Fig. 16. As stated, this design is not so easily molded as that in Fig. 15, but it is the most suitable form for gears of a comparatively wide face. No pattern dimensions are given. A drawing for machine shop use should not be confused by a maze of dimensions which are not used by the machinist. The patternmaker can be taken care of by a separate drawing, or by a special blue-print with his dimensions put on in yellow pencil.

The dimensions given are, perhaps, figured somewhat closer than is required on work of this size. The important dimensions, such as the outside diameter and the center distance, on which depend the proper fitting of the teeth of the gear, are a little too large to be measured with the vernier caliper, but they should surely be accurate within 0.005 inch—a limit easily attainable by a skillful workman.

Where a definite amount of allowable variation from the exact size can be determined on, it is customary in interchangeable manufacturing to give the dimensions with maximum and minimum limits. In work of the kind shown in Fig. 17, however, where the machine is "built" rather than "manufactured," it is not usual to give limits. The diameter of the hole in the hub of the gear is given as "6″ Std." on the



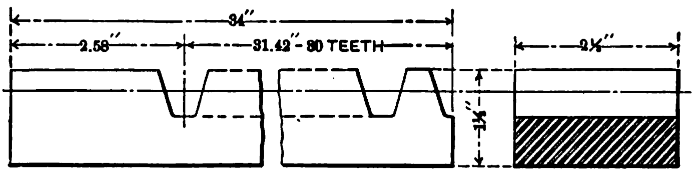
DATA FOR CUTTING

	GEAR	PINION
NUMBER OF TEETH	86	18
DIAMETRAL PITCH	2	2
WHOLE DEPTH OF TOOTH	1.0785"	1.0785"
CORRECTED ADDENDUM	0.5036"	0.5181"
CHORDAL TOOTH THICKNESS	0.7854"	0.7843"
NUMBER OF CUTTER	2	6
CENTER DISTANCE	26.000"	

Fig. 17. Example of Properly Dimensioned Gear Drawing

drawing. This means that the hole is to be bored and reamed until it will make a good push fit for a standard plug gage of the size given. It will be noted that all the dimensions needed by the workman who turns the blank, are appended to the figure, while those needed by the workman who cuts the teeth are given in tabular form.

The face view of the gear on the left is needed only for showing the number and dimensions of the arms to the pattern-maker. For pinions and webbed gears it may be omitted. It is not necessary in standard gearing to show the shape of the teeth, so the side view is given



RACK
1 WANTED - MACH. STEEL
FINISH ALL OVER

DATA FOR CUTTING

DIAMETRAL PITCH	3
WHOLE DEPTH OF TOOTH	0.7190"
ADDENDUM	0.3333"
THICKNESS OF TOOTH	0.5236"
NO. OF CUTTER	1

Machinery, N.Y.

Fig. 18. Example of Properly Dimensioned Rack Drawing

as showing the blank before the teeth are cut. The pitch and bottom circles are represented by broken and dotted circles, respectively. The shape and kind of teeth (whether involute or cycloidal) is taken care of by the cutter called for—specified by its proper number if it is involute, and by its letter if it is cycloidal.

In Fig. 18 is shown a model drawing of a rack, which is self-explanatory. Here, as in the previous case, the blank dimensions are shown attached to the figure of the rack, while the cutting dimensions are tabulated. The student may check up the dimensions given with the rules for gears and racks, page 20, if he desires practice in such calculations.

The expressions chordal tooth thickness and corrected addendum given in the table in Fig. 17, are terms which are not defined in this book. They refer to the correction of the tooth thickness and the addendum for the curvature of the pitch line—a refinement which is not commonly practiced. Full explanation of the calculations required for obtaining these dimensions have not been considered to be within the scope of this book, but will be found in "Formulas in Gearing," published by the Brown & Sharpe Mfg. Co., Providence, R. I. For ordinary work it is sufficient to simply give the addendum and tooth thickness as calculated by Rules 7, 8, 13 and 14, page 20.

CHAPTER VI

VARIATION OF THE STRENGTH OF GEAR TEETH WITH THE VELOCITY*

The generally accepted formula for calculating the strength of gear teeth is that proposed by Mr. Wilfred Lewis, first published in the Proceedings of the Engineers' Club of Philadelphia, January, 1893, and referred to in a preceding chapter.

The merit of this formula lies in the great number of variables taken into account as compared with other rules in more or less common use, and in the fact that these variables are rationally considered. The effect of each of them can be calculated with some assurance, with

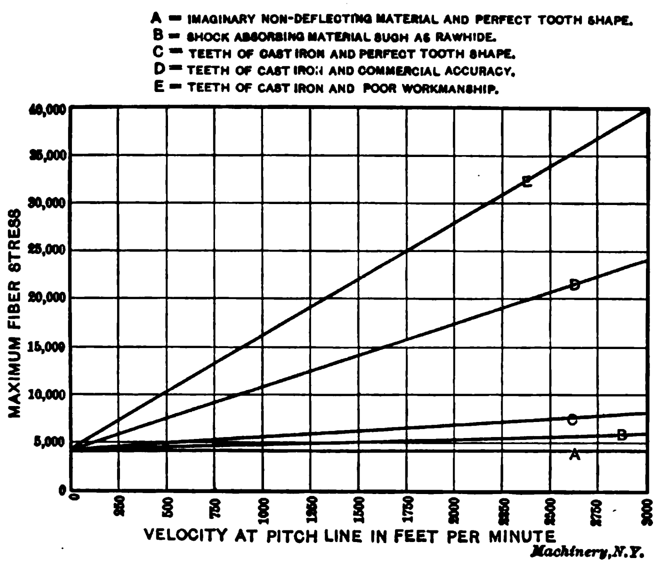


Fig. 19. Hypothetical Diagram showing the Relation of the Velocity to the Fiber Stress

the single exception of the influence of the velocity on the safe stress. In the fifteen years since the formula was first proposed, the original values for the stress as affected by the velocity have been largely used. Many designers, however, have felt that these values are rather unsatisfactory, although most of them will agree that they err rather on the side of safety than otherwise. By referring to Mr. Lewis' original paper it will be seen that these values were not given as being definitely determined, but merely as agreeing well with successful cases met with in his own practice. The following is a general analysis of the conditions involved.

* MACHINERY, January, 1908.

A variation in the strength of the teeth of a gear, due to a variation in the velocity, can be due, of course, to but one thing—impact. To illustrate this idea, and to show the cause of the impact, we will study the action of gearing under three different conditions.

1. *Gears of an imaginary undeflectable material.*—In Fig. 19 is a diagram in which the horizontal distances give velocity in feet per minute, and vertical distances give stresses in pounds per square inch, starting in this case at 4,000, which is assumed to be the maximum fiber stress in the gear we are considering, due to the load at the pitch line, which is supposed to be constant at all speeds. If the teeth of this gear are perfectly formed and well fitted together, so that there is

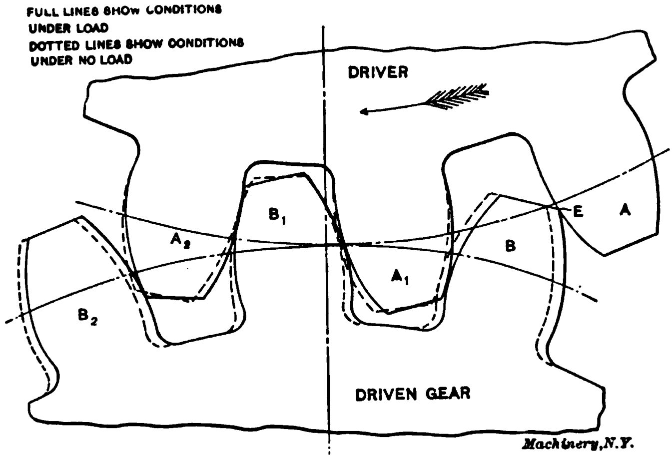


Fig. 20. The Action of Gear Teeth under Load, Greatly Exaggerated

no back lash, if the power is delivered to them steadily and smoothly, and the mechanism they drive runs without shock, any disturbance of the even movement will be impossible, and impact will be entirely absent. In the diagram in Fig. 19, then, there will be no rise of maximum fiber stresses with the velocity, so that the horizontal line *A* will show the conditions for this imaginary case.

2. *With commercial material and theoretically accurate workmanship.* The conditions in this case are shown in Fig. 20, with all the phenomena greatly exaggerated. The full lines show the conditions under load, while the dotted outlines show the conditions when the load is removed from the driven gear. The teeth A_1, B_1, and A_2, B_2, carrying the load, are deflected by it, as shown. Tooth *B*, just about to come into contact with tooth *A*, is on that account shifted from its normal position; it should be located as shown by the dotted lines. If it were in this position, it would come in contact with tooth *A* under mathematically perfect conditions, and there would be no shock of engagement. As it is, the two come suddenly into action as shown at *E*, under different conditions than those contemplated by the design, thus the contact takes place in the form of a slight blow, after which

the teeth are deflected more and more, until they have taken up their share of the load, as shown later at A_1 and B_1. If the gears are moving very slowly, the deflection takes place very slowly, and the problem is practically a static one. If the gears are running at a high velocity, the problem becomes essentially a dynamic one, and the stresses are greater than with the slow speed. The increase in stress with the increase in speed for this second case could probably be represented by a line something like C, in Fig. 19.

3. *With commercial materials and commercial accuracy.* This is, of course, the practical case to consider. A line to show the relation of the velocity to the maximum fiber stress for a given gear, would very probably look something like D in Fig. 19. This is, in fact, approximately the line which embodies the conclusions of the Lewis tables for a static stress of 4,000 pounds. It is considerably higher than line C, because impact due to irregular tooth outlines is added to the impact due to the deflection.

Practical Considerations Affecting Design

The fact that the variation of the strength with the velocity is due to impact, suggests also a number of points relating to design.

1. *Value of accuracy.* It is evident that this theory of impact puts a premium on accuracy in workmanship for gears that are to run at high speed under a heavy load. It is probable that the strength of a given pair of gears may be cut in two if the tooth outlines are not carefully determined, and if the cutter is not set centrally. This suggests the desirability of a greater sub-division of the standard cutter series for work of this kind.

2. *Resilience of design and materials.* In high-speed gearing it is evident that the shock due to the impact should be absorbed as quickly and as fully as possible. This suggests the use, at abnormally high speeds, of rawhide, wood, etc., for one of the members of the pair of gears. The introduction of spring couplings or similar devices may also be desirable, especially where the other parts of the mechanism are liable to transmit shock to the gearing.

3. *Easing off the points of the tooth.* There has always been a sort of superstition that the points of the tooth should be eased off to make the action smoother. This is done, of course, in standard involute gears, though for another reason, that of avoiding interference with the flanks of the pinions. It can now be seen that there is a solid basis for this practice in all cases where gears are to run at such speeds that severe impact is liable to take place. Referring to Fig. 20, teeth A and B are taking up the load very suddenly, owing to the fact that they are out of step, due to the deflection of the other teeth momentarily carrying the load. Easing away the points of A and B would mitigate this sudden reception of the load, allowing the inevitable deflection to take place more slowly, with a consequent gain in the strength of the gear at high speeds.

CHAPTER VII

SIMPLIFIED FORMULAS FOR STRENGTH OF GEARS*

It is generally conceded that the Lewis formula for the strength of gear teeth, with its accompanying tables, is the most accurate in form, as the maximum strength of each tooth is determined from its shape. It may be safely used for determining the strength of gears made by modern methods, but its tabulated form makes it difficult to use from the standpoint of the designer. It is well adapted to determine the strength of any given gear or pinion. But the reverse process—that of finding a gear suitable to meet the condition of a given horse-power and revolutions per minute is not so simple, the trial-and-error method being a lengthy one at the best. The following deductions give close and rapid approximations for preliminary work.

As both the gear and its pinion are usually made of the same material, either cast iron or cast steel, the strength of the pair is determined by the strength of its weakest member, which is the pinion when made of the same metal as the gear. For economical reasons the pinion is usually limited to about 15 teeth, so we may take that number as a convenient base. Circular pitch is used in the calculations, but the circular pitch can finally be transformed to diametral pitch if this is desired.

In a train of gears, the maximum reduction on any pair is usually taken at 4 or 5 to 1, so the number of reductions and ratios may be quickly deduced. Then the problem is usually presented as follows:

Given the horse-power and revolutions per minute of the pinion, what will be the allowable working stress, pitch, face, factor of strength and diameter?

The majority of trade gear lists give the horse-power of gears at 100 r. p. m. with an allowable stress for cast iron of 3,000 pounds per square inch. But it is more difficult to transform this horse-power to suit the other conditions, than to proceed independently.

The values of S, the safe working stress, which Mr. Lewis adopted tentatively, as they gave satisfactory results in practice, were as follows. (See also Table VII):

Let $V =$ speed of teeth in feet per minute and $S =$ safe working stress, then

For $V =$	100 (or less)	200	300	600	900	1,200	1,800	2,400
For cast iron: $S =$	8,000	6,000	4,800	4,000	3,000	2,400	2,000	1,700
For cast steel: $S =$	20,000	15,000	12,000	10,000	7,500	6,000	5,000	4,300

When these values are plotted, it will be seen that the curves, though slightly irregular, closely approximate curves of the hyperbolic form.

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* MACHINERY, August, 1909.

The equations of the curves which most nearly agree with the Lewis values, are found to be the following:

$$\text{For cast iron, } S = \frac{88{,}000}{\sqrt{V}}.$$

$$\text{For cast steel, } S = \frac{220{,}000}{\sqrt{V}}.$$

These formulas give the following comparative values:

When $V =$	100	200	300	600	900	1,200	1,800	2,400
For cast iron:								
$S =$	8,800	6,250	5,000	3,600	2,930	2,540	2,080	1,790
For cast steel:								
$S =$	22,000	15,625	12,500	9,000	7,325	6,350	5,200	4,475

The agreement with the Lewis assumed values is remarkably close. The new values will probably come much nearer the true ones, as they are in much better line. They are also much more dependable, as the stress suitable for any speed can be easily found from the formula to the fraction of a pound, if desired, on a true curve; whereas, the use of the tabular values results in the substitution of values which descend by variable steps of from 2,000 to 300 pounds at a jump, or if ordinary interpolation is used the result is still inaccurate, as the interpolation necessarily follows a straight line between the two nearest values, and is thus too high. The new curve values also come nearer to the comparative Harkness values as given by Kent.

The face of gears, A, is another variable quantity; but in the manufacturer's standard lists of to-day the face is usually about 3 times the pitch, and this may be adopted as close enough for preliminary work. It will be found that the majority of stock gears have either 15-degree involute or cycloidal teeth, so these styles will be used in these calculations. The factor of strength, Y', in the Lewis tables for a 15-tooth pinion of these types is 0.075. The factor Y' is found from the values of factor Y in the chart on page 29, by dividing the value of Y, as given, by 3.14. We have, therefore, the following data for a 15-tooth cast iron spur pinion:

Let $S =$ safe working stress, in pounds,
$P' =$ circular pitch, in inches,
$A =$ face, in inches,
$Y' =$ factor of strength,
$V =$ speed of pitch line, in feet per minute.

The Lewis general formula reduces to

$$\text{H. P.} = \frac{SP'AY'V}{33{,}000}$$

From our average determination above, we have:

$$S = \frac{88{,}000}{\sqrt{V}};\ A = 3\,P';\ Y' = 0.075.$$

Substituting these values in the general formula and reducing, we have for a 15-tooth cast iron spur pinion:

$$\text{H. P.} = 0.6\, P'^2 \sqrt{V} \qquad\ldots\ldots\ldots\ldots (1)$$

By a similar process, we find for a 15-tooth cast steel spur pinion:

$$\text{H. P.} = 1.5\, P'^2 \sqrt{V} \qquad\ldots\ldots\ldots\ldots (2)$$

For a bevel pinion, let

d = small diameter of bevel,

D = large diameter of bevel.

$$\text{Then H. P.} = \frac{SP'AY'V}{33{,}000} \times \frac{d}{D}.$$

As $\frac{d}{D}$ usually equals about $\frac{2}{3}$, we can say:

$$\text{H. P.} = \frac{SP'AY'V}{33{,}000} \times \frac{2}{3}$$

and for a 15-tooth cast iron bevel pinion,

$$\text{H. P.} = 0.4\, P'^2 \sqrt{V} \qquad\ldots\ldots\ldots\ldots (3)$$

For a 15-tooth cast steel bevel pinion,

$$\text{H. P.} = P'^2 \sqrt{V} \qquad\ldots\ldots\ldots\ldots (4)$$

We now wish to find V in terms of revolutions per minute. For a 15-tooth pinion, approximately:

$$V = \frac{15 \times \text{r.p m.} \times P'}{12} = 1.25\ \text{r.p.m.} \times P'.$$

Substituting this value in (1) we have:

$$\text{H. P.} = 0.6\, P'^2 \sqrt{1.25\ \text{r.p.m.} \times P'}.$$

$$\text{Squaring, H. P.}^2 = 0.36\, P'^4\, (1.25\ \text{r.p.m.} \times P').$$

Reducing, and solving for P', we have for cast iron spur pinion:

$$P' = \sqrt[5]{\frac{2.22\text{H.P.}^2}{\text{r.p.m.}}} \qquad\ldots\ldots\ldots\ldots (5)$$

A similar substitution and reduction in formulas (2), (3) and (4) gives the following:

$$\text{For cast steel spur, } P' = \sqrt[5]{\frac{0.36\ \text{H. P.}^2}{\text{r.p.m.}}} \qquad\ldots\ldots\ldots\ldots (6)$$

$$\text{For cast iron bevel, } P' = \sqrt[5]{\frac{5.0\ \text{H. P.}^2}{\text{r.p.m.}}} \qquad\ldots\ldots\ldots\ldots (7)$$

$$\text{For cast steel bevel, } P' = \sqrt[5]{\frac{0.8\ \text{H. P.}^2}{\text{r.p.m.}}} \qquad\ldots\ldots\ldots\ldots (8)$$

For rapidly varying loads, or where there is much starting and stopping, it is well to reduce the safe stress to two-thirds that allowed by the above formulas. We then have:

For cast iron spur, $\text{H.P.} = 0.4\sqrt{P'^4\,V}$; $P' = \sqrt[5]{\dfrac{5\ \text{H.P.}^2}{\text{r.p.m.}}}$ ...(9)

For cast steel spur, $\text{H.P.} = P'^2\sqrt{V}$; $P' = \sqrt[5]{\dfrac{0.8\ \text{H.P.}^2}{\text{r.p.m.}}}$ .....(10)

For cast iron bevel, $\text{H.P.} = 0.27\ P'^2\sqrt{V}$; $P' = \sqrt[5]{\dfrac{11.0\ \text{H.P.}^2}{\text{r.p.m.}}}$ (11)

For cast steel bevel, $\text{H.P.} = 0.67\ P'^2\sqrt{V}$; $P' = \sqrt[5]{\dfrac{1.8\ \text{H.P.}^2}{\text{r.p.m.}}}$ (12)

The fifth root can be easily determined by logarithms on the slide rule, or from the usual tables, but the values for the common cases are given later.

Corrections for Tooth Numbers

It now remains to determine the correction for different numbers of teeth.

As the teeth of pinions generally range from 12 to 30, we need not go outside these limits. Let N = number of teeth. Plotting the Lewis values for Y' for this case, and determining the nearest curve, we find that the straight line formula:

$$Y' = \frac{2N + 45}{1{,}000}$$

expresses this curve very closely, as will be seen by the following comparative table:

No. of Teeth, N	*Y' by Formula	Y' from Lewis' Tables	No. of Teeth, N	Y' by Formula	Y' from Lewis' Tables
12	0.069	0.067	19	0.083	0.087
13	0.071	0.070	20	0.085	0.090
14	0.073	0.072	21	0.087	0.092
15	0.075	0.075	23	0.091	0.094
16	0.077	0.077	25	0.095	0.097
17	0.079	0.080	27	0.099	0.100
18	0.081	0.083	30	0.105	0.102

Therefore, for other teeth, we can multiply the horse-power given in the above formulas by $\dfrac{2N + 45}{75}$, or more briefly by $0.027\,N + 0.6$.

Correction for Increased Velocity

We must also correct for the increased velocity of this larger pinion, *i. e.*, multiply the result by $\sqrt{\dfrac{N}{15}}$ or $0.26\sqrt{N}$. The continued product of these last two multipliers might be used, but this does not simplify the calculation. These corrections need seldom be applied for preliminary work.

To Find the Pinion Diameter

Lastly, to find the diameter of the pinion, approximately:

$$\text{diameter} = \frac{N \times P'}{\pi}, \text{ or}$$

$$\text{diameter} = 0.318\, N P',$$

or for a 15-tooth pinion,

$$\text{diameter} = 4.77\, P' \quad \text{.......... (13)}$$

If diametral pitch is desired, it is sufficiently close to say:

$$\text{diametral pitch} = \frac{3}{P'} \quad \text{.......... (14)}$$

The following formulas, therefore, Nos. (5) to (14) (as deduced above), give closely enough for all preliminary determinations, the size of pinion required of 15 teeth.

	Lewis' Tables	Stress ⅔ Lewis' Tables
Cast iron spur, $P' =$	$\sqrt[5]{\frac{2.22\ \text{H.P.}^2}{\text{r.p.m.}}}$	$\sqrt[5]{\frac{5.0\ \text{H.P.}^2}{\text{r.p.m.}}}$
Cast steel spur, $P' =$	$\sqrt[5]{\frac{0.36\ \text{H.P.}^2}{\text{r.p.m.}}}$	$\sqrt[5]{\frac{0.8\ \text{H.P.}^2}{\text{r.p.m.}}}$
Cast iron bevel, $P' =$	$\sqrt[5]{\frac{5.0\ \text{H.P.}^2}{\text{r.p.m.}}}$	$\sqrt[5]{\frac{11.0\ \text{H.P.}^2}{\text{r.p.m.}}}$
Cast steel bevel, $P' =$	$\sqrt[5]{\frac{0.8\ \text{H.P.}^2}{\text{r.p.m.}}}$	$\sqrt[5]{\frac{1.8\ \text{H.P.}^2}{\text{r.p.m.}}}$

$$\text{Diameter} = 4.77\, P'.$$

$$\text{Diametral pitch} = \frac{3}{P'}.$$

Practically, stock gears are made up to 3 inches circular pitch by ¼-inch steps, and a pitch of less than 1 inch is seldom used.

The following table will therefore determine the roots for the nearest common pitch:

No. or Root	Fifth Power	No. or Root	Fifth Power	No. or Root	Fifth Power
¾	0.24	2	32	3½	525
1	1	2¼	58	4	1,024
1¼	3	2½	98	4½	1,845
1½	8	2¾	158	5	3,125
1¾	16	3	243	6	7,776

In case the revolutions per minute of the pinion are less than 80, which is exceptionally slow, care must be taken in applying the for-

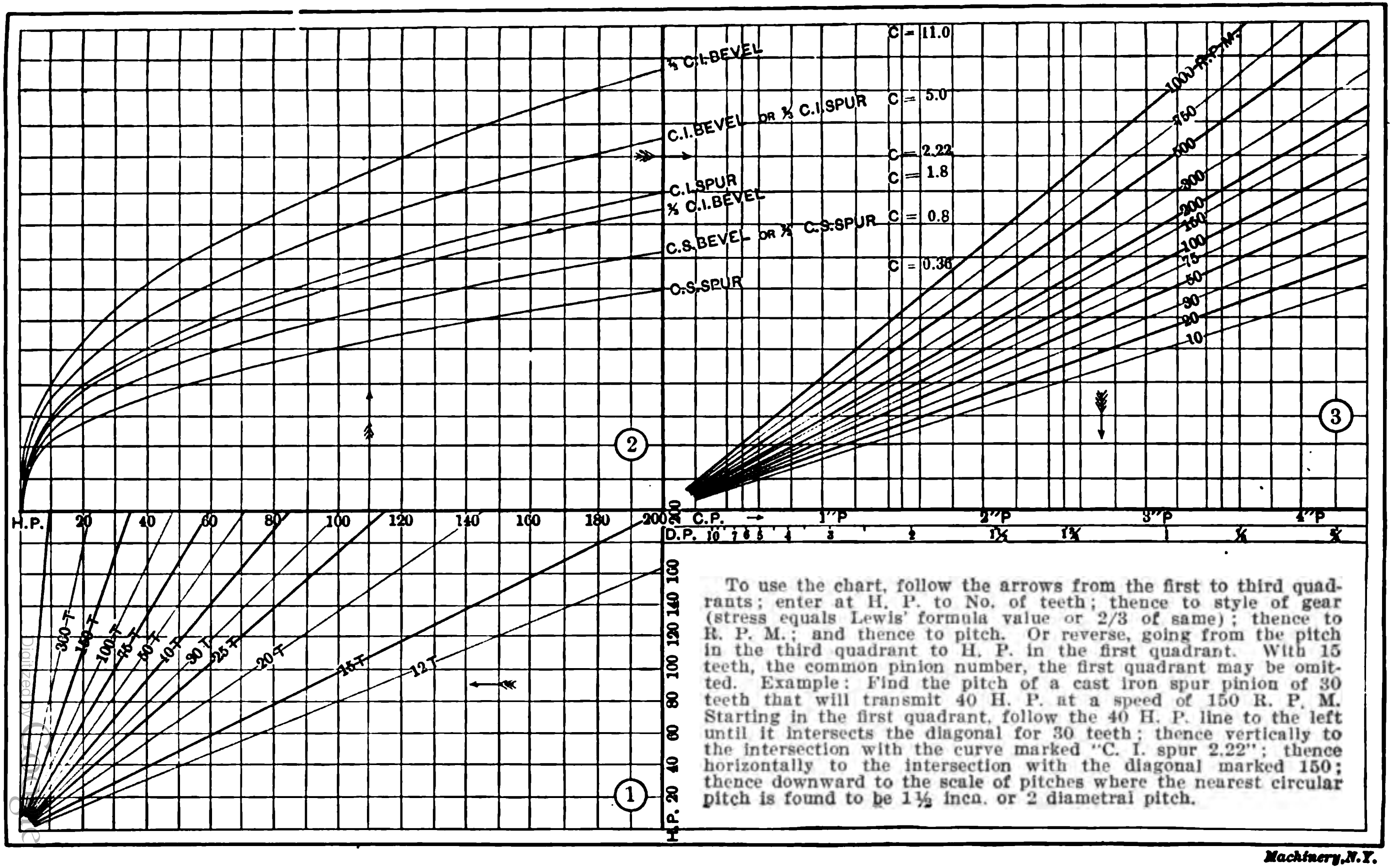

Fig. 21. Three Quadrant Gear Chart for Solution of Spur and Bevel Gear Problems, based on Lewis' Formula; for 15-degree Involute and Cycloidal Teeth

mula, or the allowable stress may be exceeded. With a 15-tooth pinion:

80 r.p.m. = 100 feet per minute for 1-inch P'.
40 r.p.m = 100 feet per minute for 2-inch P'.
27 r.p.m = 100 feet per minute for 3-inch P'.
20 r.p.m = 100 feet per minute for 4-inch P'.

Chart for Rapid Solution of Gear Problems

A simple three quadrant chart, Fig. 21, has been prepared for the rapid solution of these problems by mere inspection, good for any number of teeth, and for all the different styles, materials, and stresses of gears given by the above formulas, but for occasional preliminary determination, the formulas are sufficient, as their solution is simple.

It will, of course, be understood that the teeth considered in these formulas are those of the usual standard dimensions, in which the height of tooth equals seven-tenths of the pitch. What are known as "short tooth gears," in which the height of tooth equals half the pitch, are undoubtedly stronger, but their smaller working face is supposed to cause more rapid wear, and their use is not common. Although machine-molded cast gears run quietly at low speeds, they should not be used for rim speeds much over 1,000 feet per minute. For speeds of from 1,000 to 3,000 feet per minute cut gears should be substituted.

For a quick approximation of the diameter of the pinion shaft in inches the following formula may be used:

$$\text{Shaft diameter} = P' + 1.$$

The weight of pinions and gears varies with different makers. Pinions of from 12 to 30 teeth are usually made slightly wider than gears, even if they are not shrouded; and the smaller sizes have solid webs in place of arms. It is found that a formula of the form

$$\text{Weight in pounds} = \text{coefficient} \times P'^2AN,$$

will usually fit the weights.

For many tables, the coefficients of the following values will serve:

$$\text{Weight of pinion} = 0.35\,P'^2AN,$$
$$\text{weight of gear} = 0.45\,P'^2AN,$$

or where $A = 3\,P'$,

$$\text{Weight of pinion} = P'^3N,$$
$$\text{weight of gear} = 1.35\,P'^3N$$

or when diameter and P' are known, as $N = \dfrac{\pi D}{P'}$,

$$\text{Weight of pinion} = 3.1\,DP'^2,$$
$$\text{weight of gear} = 4.2\,DP'^2.$$

The price of gears varies largely with different manufacturers. The price of cast tooth spur gears can be usually expressed by a formula of the following form:

$$\text{Price} = (\text{coeff.} \times P'N) + (\text{coeff.} \times P').$$

Cut tooth gears usually cost about 20 per cent more than cast tooth; and cast steel gears from 50 to 75 per cent more than cast iron gears of the same size.

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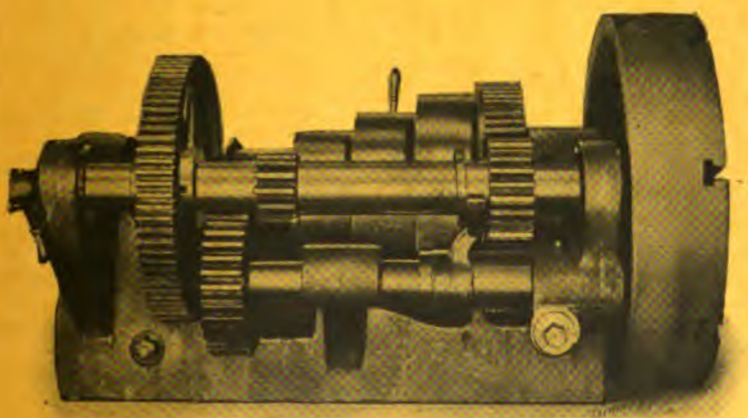

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NUMBER 16

MACHINE TOOL DRIVES

THIRD REVISED AND ENLARGED EDITION

CONTENTS

CHAPTER I

DATA FOR THE DESIGN OF DRIVING AND FEED MECHANISMS

There is probably no branch of machine design in which greater changes have taken place in recent years than that of the design of machine tools. The greater part of these changes are without doubt due to the work of Mr. Fred W. Taylor, the discoverer of high-speed steel, who has more thoroughly investigated the capabilities and possible performances of metal cutting tools than any other man. The writer had occasion some time ago to study carefully Mr. Taylor's paper "On the Art of Cutting Metals." His study of this paper, together with his own experience in machine tool design and operation, has brought him to certain conclusions in regard to some points in machine tool design which will be of interest and value not only to those who may themselves design and build such tools, but also to everyone who has to purchase or use them.

Ratio of Speed Changes

The first point to which the writer would call attention is the necessity of a sufficient number of speed changes. Those who have read Mr. Taylor's paper will remember that he shows that there is a definite relation between the cutting speed and the length of time which a tool will last without regrinding. Should the machine be run at too high a speed, the tool will last but a short time before it will have to be reground. Should it be run at too low a speed, less work, of course, will be done, although the tool will last a comparatively long time. Somewhere there is a golden mean at which the cost of machining plus the cost of tool dressing is a minimum, and theoretically our machine should always be run at that speed. Of course, in handling materials of varying grades of hardness, and, in the case of lathes and boring mills, of varying diameters, this would necessitate a very great number of speed changes. If the number of speed changes be limited, it is apparent that the machine cannot always be working at the point of maximum efficiency. The speed of cutting which gives the maximum efficiency is shown in Mr. Taylor's paper to be that speed which will destroy the tool in from 50 minutes in the case of a ⅝-inch × 1-inch roughing tool, to 1 hour and 50 minutes in the case of a 2-inch × 3-inch roughing tool. These times are of course only approximations and will vary somewhat with the cost of steel and labor and the value of the machine in which the tool is used. If the machine be slowed down from this proper speed, the cost of machining will slowly increase, but if the machine be speeded up above this proper speed, the cost of machining will increase very rapidly. In his paper Mr. Taylor gives a diagram wherein it is shown that if the machine be slowed down so that the duration of the cut is increased from 50 minutes to about 4

hours and 40 minutes, the machine is then working at about 90 per cent of its former efficiency. If the machine be speeded up until the duration of the cut is decreased to about 15 minutes, the machine will again be working at about 90 per cent efficiency. This range of speed is shown by Mr. Taylor's equations to be in the ratio of $\sqrt{15}$ to $\sqrt{280}$ or of 1 to 1.45. Consequently, if we have a machine having several speeds with the constant ratio of 1.45 between the successive speeds, we know that such a machine may always be made to operate within 90 per cent of its maximum efficiency, and that on the average it will operate at more than 95 per cent of its best efficiency.

The following table, which is derived in the manner indicated from the diagram given in Mr. Taylor's paper, shows the speed ratios corresponding to the given average and minimum efficiencies of working.

Ratio	Average Efficiency	Minimum Efficiency
1.1	99.6 per cent	99.2 per cent
1.2	98.7 per cent	97.3 per cent
1.3	97.3 per cent	94.5 per cent
1.4	95.6 per cent	91.2 per cent
1.5	93.5 per cent	87.0 per cent
1.6	90.6 per cent	81.2 per cent
1.7	86.5 per cent	73.0 per cent

From the table it will appear that even in the case of very costly machines it is of no particular advantage to reduce the ratio between successive speeds unduly. For instance, by doubling the number of speeds and reducing the speed ratio from 1.2 to 1.1, we will increase the average efficiency of the machine only about 1 per cent. It is very doubtful if the accidental variations in shop conditions would not be so great that the gain in practical work would be nothing, since the workman or the speed boss, as the case might be, would be unable to decide which of two or three speeds would be the best. The writer is therefore of the opinion that there is absolutely no practical advantage in reducing the speed ratio below 1.2 and that in the case of machines of ordinary type and cost, a ratio of 1.3 is as small as is advisable. In the case of a speed ratio of 1.3, the machine can always be made to operate at such a speed that the efficiency of working will be above 94.5 per cent and in the average case the efficiency will exceed 97.5 per cent. The 2.5 per cent loss of efficiency so caused is inappreciable as compared with other sources of loss, and it is exceedingly doubtful if the added cost of additional speed changes would not more than compensate for the possible 1 or 2 per cent of gain, entirely aside from the question of whether the extra speed changes would permit this theoretical gain to be realized.

The writer is also of the opinion that a speed ratio of more than 1.5 in the case of expensive machinery operated by highly skilled help, or of 1.7 in the case of cheap machinery operated by comparatively unskilled help is inadvisable. It will be seen that with a speed ratio of 1.5 the average efficiency of working is, in general, about 93.5 per cent, making the loss of efficiency in the average case about 6.5, or say 6 per cent. It will be seen that when the rent of the tool plus the wages of a mechanic amounts to $4 a day or upward, this 6 per cent of loss means

a money loss of $0.25 or more per day, or upward of $75 a year. Of course, an increase in the number of speed changes and reduction of ratio would not save all this loss, but assuming that it would save half of it, and further, that the machine is operating only half the time, it is evident that we can afford to spend $150 or $200 for the extra speed changes necessary in order to bring the speed ratio down to 1.3. In the case of a ratio of 1.7, the loss is 12 or 13 per cent instead of only 6 per cent, and these figures apply with greatly added force.

We are thus compelled to the conclusion that the useful range of the speed ratio in machine tool work is very narrow, ranging from 1.3 to 1.5 in ordinary cases and that a range of from 1.2 to 1.7 includes the very extremes of rational practice.

Need of Speed Changes Being Easily Made

A second point in connection with the matter of the speed changes of machine tools which is of great importance is that these changes should be easily and quickly made so that the operator will have every incentive to use the proper speed. This is a matter of less importance in the case of planers than in the case of lathes and boring mills, since a planer requires a change of speed only when the character of the material which is being cut is changed, while the lathe requires a change when any great change is made in the diameter of the work operated upon.

In this respect a motor-driven tool may have a distinct advantage over a belt-driven tool. The controller furnishes a ready means for varying the speed while the shifting of a belt from pulley to pulley is not always readily accomplished, and most machinists would much rather take two cuts of differing diameters on the back-gear than shift the belt from the small to the large pulley and throw out the back-gear in order to obtain the faster speed from the open belt. This is particularly the case when the cuts are of small duration, so that the shifting would be frequent.

It will be evident to the thoughtful mechanic that it is of great advantage to have the speed-changing mechanism so constructed that the change may be made without stopping the machine. In the case of large machines it will be of great advantage to be able to effect the speed change from the operating station, which for instance in the case of a long lathe will be the carriage. To the writer's mind the particular advantage of these refinements which he suggests, and which will be found embodied in many of the designs of our best tool makers, lies not in the fact that the time required to make the necessary speed changes is shortened, but in the fact that the workman finds it just as easy to run his machine at the proper speed as at an improper one.

Ratio of Feed Changes

A matter of even greater importance than a proper series of easily made speed changes is a proper series of easily made feed changes. A change of speed does not mean in general a correspondingly great change in the efficiency of operation of a machine tool, but a change in feed does. Mr. Taylor points out in his paper that in general the best

results in quantity of metal removed per hour are obtained when the cross-section of the chip is a maximum, even though this entails a comparatively low speed. Therefore it is of importance that the machinist be able to take the heaviest cut which the nature of his work and the power and stiffness of his machine will permit. Just as the best results in the matter of cutting speeds are obtained when the successive speeds run in geometric ratio, so the best results in the matter of feed adjustment are obtained when the successive feeds run in geometric ratio, unless the number of obtainable feeds is so great that the entire range is closely covered. For instance, a lathe equipped with the following feeds, 0.05, 0.10, 0.15, 0.20, 0.25, is distinctly inferior in productive capacity to a lathe having the same number of feeds arranged geometrically as follows, 0.05, 0.074, 0.111, 0.166, 0.25, wherein each feed is about 50 per cent greater than the preceding one.

In general the best work is obtained from a machine tool when the depth of cut is made such that the total depth of metal to be cut away is removed with one or two cuts. Such being the case, the depth of cut is practically fixed and not within the control of the operator, leaving the feed and speed as the variables which he must adjust. It is important therefore that the operator be able to take a cut as heavy as the nature of the work or of the tool will permit. Mr. Taylor's paper shows that the speed of cutting is approximately inversely proportional to the square root of the feed. It needs therefore only a very elementary knowledge of mathematics to see that if the feed must be reduced to say 80 per cent of its maximum value, the output of the lathe will be only about 90 per cent of its maximum value. Or in general, if the feed be reduced from its maximum possible value by any given per cent, then the output of the machine will be reduced from its corresponding maximum value by about one-half of that per cent. We may by means of this principle compute the ratio between successive feeds which will give us any required average value for the efficiency of operation of the machine. The values so found are tabulated below:

Efficiency	Ratio	Efficiency	Ratio
98 per cent	1.08	90 per cent	1.66
96 " "	1.18	88 " "	1.92
94 " "	1.32	86 " "	2.27
92 " "	1.46		

An inspection of the table shows that when the ratio between successive feeds is about 1.1, the average efficiency of operation of the machine may be practically perfect, and that with any considerable increase of this ratio the efficiency drops off. It is the opinion of the writer that the ratio between successive feeds should always be less than 1.3 and that, more especially in the case of expensive machinery, a value of 1.2 or less is preferable.

Importance of Convenience of Feed-changing Mechanism

It has already been pointed out that the speed-changing mechanism should be of such a character that the speed changes may be easily and quickly made. In the same way it is of even greater importance that the feed changes may be easily and quickly made. In most small

lathes which are now on the market, quick-change gears are fitted to the screw-cutting mechanism, which are equally available as quick-change gears for the feed mechanism. In most shops small lathes are not used very much of the time for screw-cutting, and in fact nine lathes out of ten are never used for that purpose, but a quick-change gear mechanism is of much greater importance when used for the purpose of obtaining feed changes than when used for the purpose of obtaining thread changes. In the average case the operator will not have to touch the thread-cutting gear once a week, while it may be advisable to change the feed every five minutes. In the case of large lathes it is advisable to have the feed changes, not in the head-stock but in the apron, in order that the workman may be encouraged to use a proper feed whenever possible.

Unlike lathes, planers are generally equipped with ratchet feeds. The successive values of the feed changes in the case of a ratchet feed will necessarily run in an arithmetic and not a geometric series, the successive feeds differing by some constant decimal of an inch. So long as the amount by which the successive feeds differ is small, and the range of feeds given by the mechanism is large, a ratchet feed is perfectly satisfactory. Many boring mills are fitted with a feed mechanism driven by a friction wheel of the type generally known as a brush wheel, the driving mechanism consisting of a steel disk of 12 to 16 inches in diameter geared to the table, and against the face of which a much smaller wheel edged with leather is pressed. It is obvious that if the steel disk rotate at a constant speed, the speed of the driven wheel and consequently the amount of the feed may be varied by adjusting its position. When it presses the disk near its center it will revolve slowly. When it presses the disk near its edge, it will revolve at a comparatively high speed. This feed mechanism has the advantage that it gives an infinite number of feed changes over a wide range, but has the disadvantage that it is not positive in its action, and lacks sufficient power for certain kinds of work. On the whole, the best feed driving mechanism is a nest of gears so arranged that any feed within the entire range may be had by the simple shifting of one or two levers.

Strength of the Feed Mechanism

In that part of his paper discussing the force required to feed the tool of a lathe or boring mill, Mr. Taylor makes the assertion that the feed mechanism should have sufficient strength to "deliver at the nose of the tool a feeding pressure equal to the entire driving pressure of the chip upon the lip surface of the tool." This would lead to the designing of a lathe or boring mill having feed gearing of equal strength with its driving mechanism. In the case of planers and other machines in which the tool is moved at a time when it is not cutting, these statements do not apply. It is not generally the custom among machine tool builders to design machines having such strong feed works as Mr. Taylor's ideas call for, and the writer sees no reason why such strength is necessary. The amount of force required to traverse a tool in a lathe is not proportional to the width of feed,

and while it may be true for fine feeds that in the case of dull tools the traversing pressure may be equal to, or greater than the downward pressure upon the tool, this is not necessarily the case with heavy feeds. As the width of the feed is increased, the downward pressure will increase almost in proportion, while the traversing pressure will increase comparatively little, so that when the lathe is taking the maximum cut which the driving mechanism is capable of handling, the pressure required to feed the tool into the work, even though it be very dull, is much less than the downward pressure. It is the writer's opinion that a feed mechanism designed to have one-half the strength of the driving mechanism is ample for large tools, while for small tools in which of course the feed will be finer, a strength of two-thirds of the driving mechanism might be preferable.

"Breaking Piece" of Feed Mechanism

The feed mechanism should be provided with a breaking piece whose strength will be less than that of the rest of the mechanism and which may be cheaply and easily replaced. The office of this piece is to prevent the breaking of the more costly and less easily replaced parts of the mechanism, exactly as the fuse in an electric circuit prevents the destruction of any other part of the circuit. Two forms of breaking piece sometimes used for such service are, first, a soft steel pin, driven through a shaft and hub of harder steel, which shears off when the strain becomes too great; and second, a short section of shaft turned down at its center, which twists off under similar circumstances. A breaking piece must be of such a character that it will not spoil any of the rest of the mechanism when it breaks, and should not cost more than a few cents, and should be as easily removed and replaced as a common change gear.

It must not be imagined that a feed gearing designed to have one-half the strength of the driving gear will not be strong enough to meet Mr. Taylor's requirements in all ordinary cases. If a tool be designed to take a maximum cut of ⅜ inch by ⅛ inch, it is not likely that much of its work will be done with such a heavy cut. If both driving and feed gearing be designed with a proper factor of safety, there is ample margin of strength for all usual conditions, while a breaking piece is the best provision against extraordinary stresses.

Pressure on Lip Surface of Tool and Its Relation to Design

The pressure upon the lip surface of the tool is required in order that the designer may know, first, the strength required of the driving mechanism and frame of a machine; second, the power required by the machine; and third, the strength required for the feed mechanism. The two materials upon which the vast majority of machine tools are called to operate are cast iron and steel. Taking first the case of cast iron, we find from Mr. Taylor's paper that the pressure upon the lip surface of the tool varies from 75,000 to 150,000 pounds per square inch of chip section in the case of soft iron, and from 120,000 to 225,000 pounds in the case of hard cast iron. The finer the feed, the greater the pressure per square inch upon the lip surface of the tool.

Thus with an ⅛-inch depth of cut and 1/64-inch feed, the pressure on the tool is about 289 pounds, or 146,000 pounds per square inch. With the same depth of cut and ⅛-inch feed, the pressure on the tool is 1,358 pounds, or only about 86,900 pounds per square inch of chip section. Both these figures are given for soft cast iron. Mr. Taylor gives formulas for the total pressure of the work upon the lip surface of the tool, but the following table will be found more convenient for obtaining the required values, although the figures given are of course only approximations:

	Pressure per Square Inch	
Feed, Inches	Soft Cast Iron	Hard Cast Iron
1/64	140,000	220,000
1/32	120,000	190,000
1/16	100,000	160,000
1/8	85,000	135,000

In the case of soft and medium steels we find that the pressure in pounds per square inch of chip section runs from 250,000 to 300,000 pounds, being greater in the case of the finer feeds. In the case of special steels which combine high tensile strength and great elongation, it is probable that these figures would be very much exceeded. The amount of the feed and depth of cut will depend on the kind of work which is to be machined. In the case of small castings, 3/16 inch is an ample allowance for depth of cut and ⅛ inch would be much more usual. In the case of very large and heavy castings the depth of cut required might run up to ½ inch, and in the case of large "meaty" forgings, it may be even greater than this at some places. In those cases where the area of chip section is not fixed by the work, as in the case of stocky forgings and castings, the greatest width of feed is limited by the strength of the machine itself, which in turn is limited only by the length of the purchaser's purse. Presumably it would be possible to build a boring mill or a planer capable of taking a cut an inch deep with an inch feed if anyone wished to pay for such a machine, but whether it could do the average line of work as economically as a machine taking a ⅜-inch cut with ⅛-inch width of feed is another matter. While there is no settled rule either for the maximum depth of cut or width of feed for any particular type of machine, the matter of the size of tool used is generally definitely known. In the case of forged roughing tools the maximum chip section will be from 2 to 3 per cent of the area of the section of the tool shank. For instance, the heaviest cut which a tool forged from 1-inch by 1½-inch stock will be called upon to take will be ¼ inch by ⅛ inch, or perhaps a trifle greater. In the case of tools ground from bar stock and held in tool-holders, the section of the chip may run up as high as 5 per cent of the section of the bar. Knowing the size of tool for which the tool-holders are designed, we may proportion our machine accordingly.

A matter which has great effect not only upon the quantity of work which a machine is capable of doing, but also upon its accuracy and length of useful life, is its stiffness. While it is true that if we know the maximum pressure upon the lip-surface of the tool, we may design

a machine for strength and have one which will probably never break in service, yet it is often better to add many times the quantity of metal which mere strength would call for, in order to have a machine with the maximum of stiffness. Stiffness in machine tool design has to do with two points, the first being the actual deflection of the metal of which it is composed under the stresses which come upon it in operation; the second is the play which invariably exists at all joints, more especially the slides of compound rests in lathes, and of saddles in boring mills and planers. The best remedy for actual deflection of metal is to use plenty of it, and to distribute it in such a way as to realize from it its maximum strength. The writer has found that an excellent method of designing such machine parts as require great stiffness is by comparison with existing tools whose operation is satisfactory. Let us assume for instance that we are to design the cross-rail of a planer. The rail is to be 8 feet between the housings and the overhang of the tool below the center of the rail is to be 30 inches. The cut is to be, let us say, ½ inch deep by ⅛ inch feed. Let us assume further that we have at our disposal a 4-foot planer, the overhang of whose cutting tool is 15 inches, and which will take in a satisfactory manner a cut ¼ inch deep by 1/16 inch feed. We now have sufficient data to satisfactorily design a cross-rail for the larger planer. If we assume that the deflection of the tool produced in the two cases should be identical in order to have the work equally satisfactory, we will find that the pressure upon the tool of the larger planer will be 4 times that upon the tool of the smaller; that both the bending and the twisting moments set up in the cross-rail will be 8 times as large, and that the distance over which these moments will operate to produce a deflection will be twice as great. Therefore, if the two rails had the same cross-section, the deflection of the tool of the larger machine would be 16 times that of the tool of the smaller. The stiffness of two bodies of similar section varies directly as the 4th power of the ratio of their homologous dimensions. Therefore, if we make the section of the rail of the larger machine similar in form to that of the rail of the smaller machine, each dimension twice as great as the corresponding dimension of the smaller rail, it will be 16 times as stiff and the deflections in the two cases will be identical. In case the rail of the smaller machine were not of the best form to resist the stresses which it must sustain, the form might be changed, the designer using his best judgment as to what effect such change might have upon its stiffness.

CHAPTER II

SPEEDS AND FEEDS OF MACHINE TOOLS

In designing machine tools of any type, be it a lathe, milling machine, grinding machine, etc., aside from the correct proportioning of the parts, and the introduction of convenient means for rapidly producing certain motions, a very important factor is to be taken into consideration, that is, the correct proportioning of the speeds and feeds of these various machines. Before entering into an explanation of the method which is to be set forth later, we will explain some of the preliminary considerations which are to be met by the designer. Supposing a problem of designing a lathe be presented; it follows, at once, that certain conditions limiting the problem are also given. These limiting conditions may be considered as the size and material of the piece to be turned.

We consider the material of a piece to be machined as a limiting condition for the reason that a lathe turning wood must run at a different speed from one turning brass, and the latter at a different speed from a lathe turning iron or steel. Then, again, in turning a small piece, our machine will revolve faster than in turning a large piece. The speeds required for machining advantageously the different materials, according to the different diameters, may be termed "surface speeds." Roughly speaking, the surface speeds for the different materials vary within comparatively narrow limits. We may assume the following speeds for the following materials (using carbon steel cutting tools):

Cast iron	30 to 45 feet per minute.
Steel	20 to 25 feet per minute.
Wrought iron	30 feet per minute.
Brass	40 to 60 feet per minute.

For cast iron as found in Europe, we may assume 20 to 35 feet per minute; this lower figure is due to the fact that European cast iron is considerably harder.

The surface speeds above given are, of course, approximate, and it is left to the judgment of the designer to modify them according to the special given conditions. These surface speeds for cutting metal are the same whether the piece to be cut revolves, or the cutting tool revolves around the piece, or, as in a planer, the cutting tool moves in a straight line along or over the work. Therefore, the surface speeds in a general sense hold good for all types of machines, such as milling machines, lathes, gear-cutting machines, drilling machines, planers, etc.

Suppose that a problem is given requiring that a lathe be designed to turn both cast iron and steel, and to turn pieces from one-half inch

to twelve inches in diameter. Simple calculation will show us that a piece of work one-half inch in diameter, and having a surface speed of 30 feet per minute, as would be suitable for cast iron, must make 230 revolutions per minute. A piece of steel, which is 12 inches in diameter, with a surface speed of 20 feet per minute, must make 6.5 revolutions per minute approximately. It follows that the lathe to conform to the conditions imposed, must have speeds of the spindle varying from 6.5 to 230 revolutions per minute. These are the maximum and minimum speeds required. To meet the varying conditions of intermediate diameters, the lathe will be constructed to give a certain number of speeds. The lathe, probably, will be back-geared and have a four-, five-, or six-step cone.

In a correct design these various speeds must have a fixed relation to each other. For reasons explained in Chapter III these speeds must form a geometrical progression, and the problem briefly stated is this: "The speeds (the slowest and fastest being given) are to be proportioned in such a manner that they will form a geometrical progression." The ratio of the gearing is also to be found. A geometrical progression in a series of numbers is a progressive increase or decrease in each successive number by the same multiplier or divisor at each step, as 3, 9, 27, 81, etc.

To treat the problem algebraically let there be

$n =$ number of required speeds,
$a =$ slowest speed,
$b =$ fastest speed,
$d =$ number of speeds of cone,
$n-1 =$ number of stops or intervals in the progression of required speeds,
$f =$ ratio of geometrical progression, or factor wherewith to multiply any speed to get the next higher.

Algebraically expressed, the various speeds, therefore, form the following series:

$$a, af, af^2, af^3 \ldots\ldots. af^{n-2}, af^{n-1}$$

The last, or fastest speed, is expressed by af^{n-1} and also by the letter b. Therefore, $af^{n-1} = b$, or

$$f^{n-1} = \frac{b}{a}, \text{ and } f = \sqrt[n-1]{\frac{b}{a}}$$

Suppose we have, as an example, a lathe with a four-speed cone, triple geared. In this case we would have four speeds for the cone, four more speeds for the cone with back-gears, and still four more speeds with triple-gears; therefore, in all, twelve speeds. Assuming a as the slowest speed in this case, b would be expressed by af^{11}, and the series, therefore, beginning with the fastest speed, would run

$$af^{11}, af^{10}, af^9 \ldots\ldots.. af^2, af, a.$$

The four fastest speeds, which are obtainable by means of the cone alone would be

$$af^{11}, af^{10}, af^9, af^8.$$

Dividing each of the four members of this series by f^4, we obtain the following series:

$$af^7,\ af^6,\ af^5,\ af^4,$$

as the speeds of cone with back-gears.

Again dividing the series of speeds of the cone af^{11} to af^8 by $f^4 \times f^4 = f^8$ we obtain the series

$$af^3,\ af^2,\ af,\ a,$$

as the series of speeds of cone with triple-gears.

We have, therefore, in this way accounted for all the twelve speeds that the combination given is capable of, and it is now very evident that the ratio of the back-gears must be f^4, or, in general, f^d, if $d =$ number of speeds of cone, and the ratio of triple-gears f^8 (or, in general, f^{2d}).

By carrying this example still further, we would find that the ratio of quadruple-gears would be f^{3d}.

We can summarize the preceding statements, and put them in a more convenient form for calculation by writing:

$$\begin{aligned}
lg \text{ of ratio of back-gears} &= d\, lg\, f\\
lg \text{ of ratio of triple-gears} &= 2d\, lg\, f\\
lg \text{ of ratio of quadruple-gears} &= 3d\, lg\, f
\end{aligned}$$

The problem, with this consideration, therefore, is solved. An example will be worked out below.

We will now consider a complication of the problem which very often occurs. Should the overhead work of the drive in consideration have two speeds, then we will obtain double the number of available speeds for the machine, and this number of speeds may be expressed by $2n$, in order to conform to the nomenclature used above. This modified problem is treated just as the problem above, and the series of speeds is found as in the first case, and we have as a factor

$$f = \sqrt[2n-1]{\frac{b}{a}}$$

We must consider now that one-half the obtained speeds are due to the first overhead speed, the other half to the second.

In writing the odd numbers of speeds found in one line, and the even numbers of speeds in another, we obtain the following two series:

$$\begin{aligned}
&a,\ af^2,\ af^4 \ldots\ldots\ldots\ldots af^{2n-4},\ af^{2n-2}\\
&af,\ af^3,\ af^5 \ldots\ldots\ldots\ldots af^{2n-3},\ af^{2n-1}
\end{aligned}$$

In examining these two series, we will find that they are both geometrical progressions, and furthermore, that both progressions have the same factor, and calling this factor, f_1, we have

$$f_1 = f^2,$$

and the ratio of the two counter-shaft speeds is equal to f, because to obtain any speed in the second series we multiply the corresponding speed in the first series by f. The two series in our case are due to the two overhead speeds. We need to concern ourselves with only one (either one of the two series), and without going again through the

explanation for the first case, it is very evident that we will arrive at the following conclusions:

$$lg \text{ of ratio of back-gears} = d\, lg\, f_1$$
$$lg \text{ of ratio of triple-gears} = 2d\, lg\, f_1$$
$$lg \text{ of ratio of quadruple-gears} = 3d\, lg\, f_1$$

Having in this way obtained all the desired speeds and the ratios of the gears, it is a simple matter for the designer to determine the actual diameters of the various steps for the cone and for the gears. To do so he has at his disposal various methods,* which need not be explained here. The main thing for him to have is a geometrical progression of speeds, as a foundation for his design.

Problem 1. A Triple-Geared Lathe

Suppose the following example to be given: Proportion the speeds and find the gear ratio of a six-step cone, triple-geared lathe; slowest speed, 0.75 revolution per minute; fastest, 117 revolutions per minute.

This example of a six-step cone, triple-geared, will give us eighteen available speeds. Using our previous notation, $n = 18$, $n - 1 = 17$, $a = 0.75$, and $b = 117$; therefore

$$f = \sqrt[17]{\frac{117}{0.75}} = \sqrt[17]{156}$$

The slowest speed being given, we multiply it by the factor f to obtain the next higher, and this one in turn is again multiplied by the

COMPLETE CALCULATION OF CONE PULLEY SPEEDS

$lg\ 0.75 = 0.8750613 - 1$
$lg\ f = 0.1290073$
$0.0040686 = lg\ 1.009$
0.1290073
$0.1330759 = lg\ 1.358$
0.1290073
$0.2620832 = lg\ 1.828$
0.1290073
$0.3910905 = lg\ 2.461$
0.1290073
$0.5200978 = lg\ 3.312$
0.1290073
$0.6491051 = lg\ 4.457$
0.1290073
$0.7781124 = lg\ 5.999$
0.1290073
$0.9071197 = lg\ 8.074$
0.1290073

$1.0361270 = lg\ 10.867$
0.1290073
$1.1651343 = lg\ 14.626$
0.1290073
$1.2941416 = lg\ 19.685$
0.1290073
$1.4231489 = lg\ 26.494$
0.1290073
$1.5521562 = lg\ 35.658$
0.1290073
$1.6811635 = lg\ 47.991$
0.1290073
$1.8101708 = lg\ 64.591$
0.1290073
$1.9391781 = lg\ 86.932$
0.1290073
$2.0681854 = lg\ 117.000$

* See MACHINERY'S Reference Series, No. 14, Details of Machine Tool Design, Chapters I and II.

factor f, and so on, until we have reached the highest speed b. The 17th root of 156 is easiest found by the use of logarithms.

We have

$$lg\,156 = 2.1931246$$
$$lg\,f = 1/17\;lg\,156 = 0.1290073$$
$$f = 1.3459$$

Now we follow out the multiplication by finding the logarithm of 0.75, the slowest speed, adding to it the logarithm of the factor f to obtain the logarithm of the next higher speed; and adding the logarithm of factor f to the sum of these two logarithms will give us the logarithm of the next higher speed. By looking up the numbers for these logarithms, we find these speeds to be 1.009 and 1.358. The complete calculation is given in tabulated form on the previous page.

Now, for example, the number of speeds of cone d equals 6, and according to our formula, the logarithm of the ratio of the back-gears $= d\;lg\;f$, and the logarithm of the ratio of the triple-gears $= 2d\;lg\;f$. Expressed in figures we have:

$lg\,f = 0.1290073 \times 6 = 0.7740438$, and the ratio of the back-gears $=$ 5.9435. Further, $12\;lg\,f = 1.5480876$, and the ratio of the triple-gears $=$ 35.325.

Problem 2.—Lathe with two Counter-shaft Speeds

Suppose the following example is given: Proportion the speeds and find the gear-ratio of a four-step cone, back-geared, two speeds to counter-shaft; slowest speed, 25 revolutions per minute; fastest speed, 500 revolutions per minute.

In this case $n = 8$; $2n = 16$; and, consequently,

$$f = \sqrt[15]{\frac{500}{25}} = \sqrt[15]{20} = 1.221$$

In following out the calculation as shown in Problem 1, we obtain the following series of sixteen speeds:

1) 25.00	5) 55.58	9) 123.54	13) 274.64
2) 30.53	6) 67.86	10) 150.85	14) 335.35
3) 37.28	7) 82.86	11) 184.20	15) 409.48
4) 45.51	8) 101.18	12) 224.92	16) 500.00

Of these sixteen speeds, eight are due to one over-head work speed; the other eight are due to the second over-head work speed. We write the odd and even speeds in two series, as below:

First Series.	Second Series.
1) 25.00	2) 30.53
3) 37.28	4) 45.51
5) 55.58	6) 67.86
7) 82.86	8) 101.18
9) 123.54	10) 150.85
11) 184.20	12) 224.92
13) 274.64	14) 335.35
15) 409.48	16) 500.00

In order to find the ratio of the back-gears, we can use either one of these two series, and as explained above, $f_1 = f^2$. We therefore

have $1.221^3 = f_1$, and further $4 \times lg\, f_1 =$ ratio of back-gears. From this the ratio of the back-gears = 4.9418. We also know that the ratio of counter-shaft speeds $= f = 1.221$.

This method of geometrically proportioning speeds in machine drives, which has been explained at length, will be found, after one or two applications, a rather simple one. But its usefulness is not limited to the proportioning of speeds in machine drives, as it can also be applied to the proportioning of feeds.

Feeds for Machine Tools

Before proceeding to apply this method to geometrically proportioning feeds in machines, a few remarks on feeds may not be out of place. By feeds are understood the advances of table, carriage, or work, in relation to the revolutions of the machine spindle. Feeds may be expressed in inches per minute or inches per revolution of spindle. In a table given below, feeds for different machines are given in inches for one revolution per spindle, where not otherwise specified. This table is supposed to represent modern practice, with carbon steel cutting tools, but the figures given, of course, represent general experience, and special cases, no doubt, will often modify them considerably.

	Feed, Inches.
Plain milling machine	0.005 – 0.2
Large plain milling machine	0.010 – 0.3
Universal milling machine	0.003 – 0.2
Large universal milling machine	0.003 – 0.25
Automatic gear cutter, small	0.005 – 0.1
Drills (spindle-feed)	0.004 – 0.02
Planing machine (traverse feed)	0.005 – 0.7
Slotting machine (feed of work)	0.005 – 0.2
Drilling long holes in spindles (per revolution of drill)	0.003 – 0.01
Lathes, feed for roughing	56 – 80 turns per inch
Lathes, feed for finishing	112 turns per inch.

Universal Grinding Machine

Surface speed of emery-wheel, 4,000–7,000 feet per minute. Traverse of platen or wheel, 2 to 32 inches per minute; the fast feeds are for cast iron. Surface speed of work on centers, 130–160 feet per minute. For internal work use the following surface speeds of emery-wheel (highest nominal speeds), with no allowance for slip of belt; lowest nominal speed about 40 per cent less. Any speed between should be obtainable.

Diameter of Wheel.	Feet per Minute.
1 5/8	3,600
1	2,750
3/4	2,100
7/16	1,450
1/4	1,100

Surface Grinding Machine

Surface speed of emery wheel, 4,000–7,000 feet per minute. Table

speed per minute, 8–15 feet. Cross feed to one traverse of platen, 0.005–0.2 inch. Cross feed to one revolution of hand-wheel, 0.25 inch.

Problem 3 — The Feeds of a Milling Machine

The problem of proportioning the feeds of different machines varies in each case, although always embodying similar principles. It is, therefore, proposed to take a typical case and apply the method to the problem presented, and in this way explain the advantages of the particular method referred to.

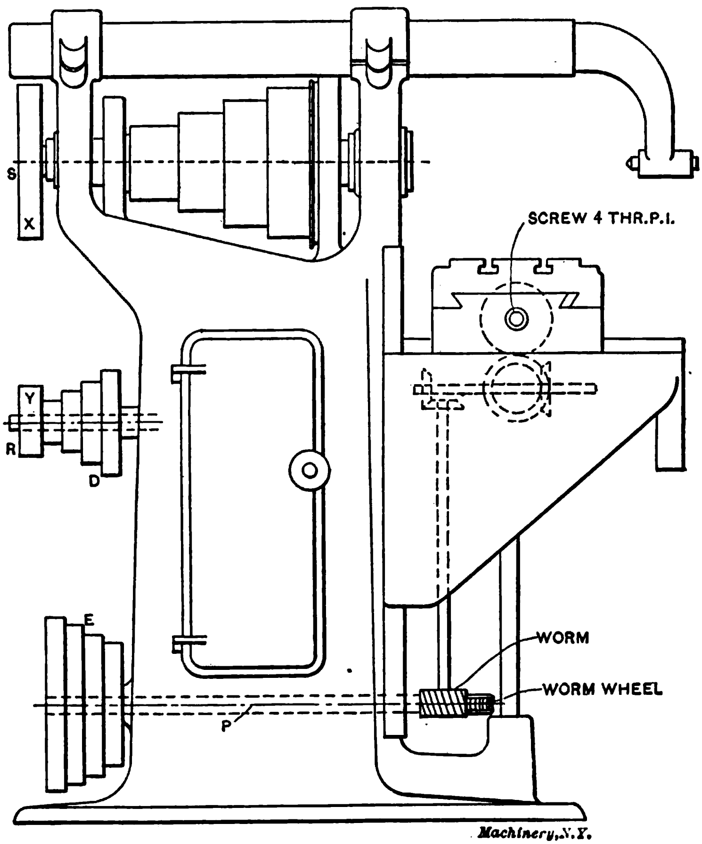


Fig. 1. General View of Milling Machine, having Cone Pulley Feed

In Fig. 1 is given an outline drawing of a milling machine. The type selected is not one of the latest designs, because it is easier to comprehend the principles involved in a type such as shown. The application of the principles, however, is, with few modifications, the same for the most modern gear-feed types, as for the one shown. The problem in this case will be the following: Given the fastest and slowest feeds per one revolution of main spindle, proportion the

required feeds in such a manner that they will form a geometrical progression. Cones D and E as well as pulleys X and Y can be transposed.

The main data with which we have to concern ourselves about this machine may be assumed to be as follows: lead screw, four threads per inch, single; advance of screw per one revolution, 0.25 inch; largest feed wanted, 0.25 (equal to one revolution of screw); smallest feed wanted, 0.005 inch (equal to 1/50 revolution of screw); for one revolution of screw, shaft P (see Fig. 1) makes thirty revolutions; for 1/50 revolution of screw, shaft P makes $30 \div 50 = 0.6$ revolutions. The ratio of revolutions between the screw and shaft P is therefore in our example as 1 to 30; that is, given the revolutions of shaft P we divide this number by 30 to obtain the revolutions of the screw. The revolutions of the screw multiplied by the lead L (in this case equal to 0.25) gives the advance for given revolutions of P. Let

V = ratio of train from P to screw,

L = lead of screw,

R_p = revolutions of shaft P per one revolution of spindle,

p = advance or feed of screw per one revolution of spindle, expressed in inches.

We have

$$p = \frac{R_p L}{V} \qquad (1)$$

$$R_p = \frac{Vp}{L} \qquad (2)$$

If now n equals the numbers of feeds wanted, we obtain for f, the factor wherewith to multiply each feed to get the next higher feed,

$$f = \sqrt[n-1]{\frac{b}{a}}$$

in which b is the fastest, and a, the slowest speed of shaft P. That is, in the present case

$$R_p \text{ maximum} = 30 = b.$$
$$R_p \text{ minimum} = 0.6 = a.$$

The problem in our case stated that cones D and E, as well as pulleys X and Y could be transposed. The cones have four steps, and transposing them gives us eight speeds. Pulleys X and Y being also transposable gives, therefore, $2 \times 8 = 16$ speeds. The numerical value for f is therefore in our case,

$$f = \sqrt[15]{\frac{30}{0.6}} = \sqrt[15]{50}$$

The maximum and the minimum speeds of shaft P per one revolution of spindle of machine, as well as the number of steps required, being known, we now readily obtain a geometrical series with the minimum speed of shaft P as a beginning, and the maximum speed as the

last step. The numerical values that follow are found exactly in the same way as the values for the different speeds of a lathe drive as already shown. The required speeds of shaft P are then:

1) 0.6	5) 1.70	9) 4.83	13) 13.72
2) 0.78	6) 2.21	10) 6.27	14) 17.81
3) 1.01	7) 2.87	11) 8.14	15) 23.11
4) 1.31	8) 3.72	12) 10.57	16) 30.00

The value of p, in our case, becomes, according to formula (1),

$$p = \frac{R_p \times 0.25}{80} = 0.0083\, R_p$$

in which R_p, the number of revolutions of shaft P, has the different values found above. By substituting these values of R_p, we obtain the following feeds, which are the feeds of the lead screw per one turn of machine spindle.

1) $0.6 \times 0.0083 = 0.005$ inches	9) $4.83 \times 0.0083 = 0.0400$ inches
2) $0.78 \times 0.0083 = 0.0065$ "	10) $6.27 \times 0.0083 = 0.0520$ "
3) $1.01 \times 0.0083 = 0.0084$ "	11) $8.14 \times 0.0083 = 0.0677$ "
4) $1.31 \times 0.0083 = 0.0109$ "	12) $10.57 \times 0.0083 = 0.0877$ "
5) $1.70 \times 0.0083 = 0.0141$ "	13) $13.72 \times 0.0083 = 0.1138$ "
6) $2.21 \times 0.0083 = 0.0183$ "	14) $17.81 \times 0.0083 = 0.1513$ "
7) $2.87 \times 0.0083 = 0.0238$ "	15) $23.11 \times 0.0083 = 0.1918$ "
8) $3.72 \times 0.0083 = 0.0308$ "	16) $30.00 \times 0.0083 = 0.2500$ "

We now write the speeds found for shaft P in two columns, one containing the odd numbers and the other the even numbers, in this manner:

1) 0.6	2) 0.78
3) 1.01	4) 1.31
5) 1.70	6) 2.21
7) 2.87	8) 3.72
9) 4.83	10) 6.27
11) 8.14	12) 10.57
13) 13.72	14) 17.81
15) 23.11	16) 30.00

The series of speeds in each column forms a geometrical progression, and we assume that the speeds in the first column are due to the position of the pulleys X and Y as shown in the outline drawing, Fig. 1, and that the speeds in the second column are due to a reversed position of X and Y. That is to say, the speeds in the second column above are obtained after having changed Y to X and X to Y. As these speeds in the second column are equal to the speeds in the first column multiplied by factor f, it follows that the two speeds of shaft R are to each other as 1 is to f. Assuming these two speeds to be m and n, the proportion exists,

$$m : n = 1 : f \tag{3}$$

Supposing x and y to represent the diameters of the respective pulleys; it will be evident that

$$1 \times x = my\text{; or, } m = \frac{x}{y} \tag{4}$$

$$1 \times y = nx\text{; or, } n = \frac{y}{x} \tag{5}$$

Substituting the values (4) and (5) in formula (3) we have

$$\frac{x}{y} : \frac{y}{x} = 1 : f\text{, or } f = \frac{y}{x} : \frac{x}{y} = \frac{y}{x} \times \frac{y}{x} = \frac{y^2}{x^2} \tag{6}$$

The value of f being known, we have in formula (6) an expression of the relation which the diameters of the pulleys X and Y must bear to each other. Putting this formula into a more handy shape we find from $f = \dfrac{y^2}{x^2}$

$$y^2 = f x^2\text{, or } y = \sqrt{f x^2} \tag{7}$$

$$x^2 = \frac{y^2}{f}\text{, or } x = \sqrt{\frac{y^2}{f}} \tag{8}$$

In using either (7) or (8), and assuming one diameter, the other one is easily found. The remaining part of the problem, that is, to find the diameters of the cone, is now a simple matter.

CHAPTER III

MACHINE TOOL DRIVES

The present chapter contains considerable matter already treated in Chapter II. In order to make the present chapter a complete whole by itself, it has, however, been considered advisable to repeat such statements and formulas as are necessary to fully comprehend the somewhat different treatment of the subject presented in this chapter.

One of the first problems encountered in the design of a new machine tool is that of laying out the drive. The importance of a properly proportioned drive is coming more and more to be recognized. The use of high-speed steels, and the extra high pressure under which modern manufacturing is carried on, precludes the use of any but the most modern and efficient drive.

The drive selected may be one of the following different kinds, depending on the conditions surrounding the case in hand: We may make the drive to consist of cone pulleys only; we may use cone pulleys in conjunction with one or more sets of gears; or we may make our drive to consist of gears only, depending on one pulley, which runs at a constant speed, for our power. If the conditions will allow, we may use an electric motor, either independently or in connection with suitable gearing.

After having selected the form which our drive is to take and the amount of power to be delivered, which we will assume has been decided upon, we may turn our energies to the problem of arranging the successive speeds at which our machine is to be driven. As most machines requiring the kind of drive with which we are here concerned have spindles which either revolve the work, or a cutting tool that has to be worked at certain predetermined speeds dependent on the peripheral speed of the work or cutter, a natural question to be asked at this point is, "What is the law governing the progression of these speeds?"

As an example to show what relation these speeds must bear to one another, let us suppose that we have five pieces of work to turn in a lathe, their diameters being 1, 2, 5, 10, and 20 inches respectively. In order that the surface speed may be the same in each case we must revolve the one-inch piece twice as fast as the two-inch piece, because the circumference varies directly as the diameter, so that a two-inch piece would be twice as great in circumference as the one-inch piece. The five-inch piece would revolve only one-fifth as fast as the one-inch piece; the 10-inch piece 1/10th, the 20-inch piece 1/20th. We have seen that the addition of one inch to the diameter of the one-inch piece reduces the speed 100 per cent. If we add one inch to the two-inch piece we reduce the speed 50 per cent, and similarly one inch added to the 5-, 10-, and 20-inch pieces reduces the speed 20, 10, and 5 per cent respectively. From this we see that the speed must vary inversely with the diameter for any given surface speed. It also shows that the speeds differ by small increments at the slow speeds, the increment gradually increasing as the speed increases. Speeds laid out in accordance with the rules of geometrical progression fulfill the requirements of the above conditions.

If we multiply a number by a multiplier, then multiply the product by the same multiplier, and continue the operation a definite number of times, we have in the products obtained a series of numbers which are said to be in geometrical progression. Thus 1, 2, 4, 8, 16, 32, 64 are in geometrical progression, since each number is equal to the one preceding, multiplied by 2, which is called the ratio. The above may be expressed algebraically by the following formula:

$$b = a\,r^{n-1}$$

where b is a term or number which is the nth term from a which i the first term in the series. The term r is the ratio or constant multiplier.

If we are given the maximum and minimum of a range of speeds we may find the ratio by the following formula, when the number of speeds is given:

$$r = \sqrt[n-1]{\frac{b}{a}}$$

As most cases in which we would use this formula would require the use of logarithms, we will express the above as

$$\text{Log}\, r = \frac{\text{Log}\, b - \text{Log}\, a}{n - 1}$$

Let us suppose we are designing a drive which is to give a range of 18 spindle speeds, from 10 to 223 revolutions per minute. Now the first thing to be done is to find the ratio r, which, by the above formula is found to be 1.20, and by continued multiplication, the series is found to be 10, 12, 14.4, 17.25, 20.7, 24.85, 29.8, 35.8, 43, 51.6, 62, 74.4, 89.4, 107, 129, 155, 186, 223.

Our drive can be made to consist of one of the many forms just mentioned. As the cone and back-gear is the most common form, and fills the conditions well, we will choose that style drive for the case in hand. We may have a cone of six steps, double back-gears and one counter-shaft speed, such as would be used in lathe designs, or we may use a cone with three steps, double back-gears and two counter-shaft speeds as is used in milling machines. This latter plan will be followed in our present case.

There are two methods of arranging the counter-shaft speeds. First, by shifting the machine belt over the entire range of the cone before changing the counter-shaft speed; and second, by changing the counter-shaft speed after each shift of the machine belt. The method used will have a very important effect on the design of the cone. The cone resulting from the former practice will be quite "flat," with very small difference in the diameter of the steps, while the use of the second method will produce a cone which will have a steep incline of diameters. Some favor one, some the other. The controlling point in favor of the first method is the appearance of the cone obtained.

We will first design our drive with the conditions of the first method in view; that is, we will arrange our counter-shaft speeds so that the full range of the cone is covered before changing the counter-shaft speed, thus obtaining the flat cone. Tabulating the speeds in respect to the way they are obtained, we have

Cone.	Open Belt.		Small Ratio Back Gears in.		Large Ratio Back Gears in.	
	Fast Counter.	Slow Counter.	Fast Counter.	Slow Counter.	Fast Counter.	Slow Counter.
Step 1.....	223	129.	74.4	43.	24.85	14.4
Step 2.....	186	107.	62	35.8	20 7	12.
Step 3.....	155	89.4	51.6	29.8	17.25	10.
	1	2	3	4	5	6

From the above table we may obtain the ratio of the two sets of back-gears, the counter-shaft speeds, and the speeds off each step of the cone.

The ratio of the large ratio back-gears is found by dividing one term in column 2 by a corresponding term in column 6. The ratio of the small ratio gears is found by dividing a term in column 2 by a corre-

sponding term in column 4. The ratio of counter-shaft speeds is obtained by dividing a term in column 5 by a corresponding term in column 6; and the ratio of speeds off each step of the cone, by dividing the term corresponding to step 1 in any column by a term corresponding to step 2 or 3, as desired, from the same column. The results for the present case are as follows:

Ratio of large ratio gears is........................	8.94 to 1
Ratio of small ratio gears is........................	2.98 to 1
Ratio of counter-shaft speeds is....................	1.725 to 1
Ratio of speeds off step 1 to those off step 2........	1.2 to 1
Ratio of speeds off step 1 to those off step 3........	1.44 to 1

The matter of designing the cone seems to cause trouble for a good many, if we are to judge by the results obtained, which are various in

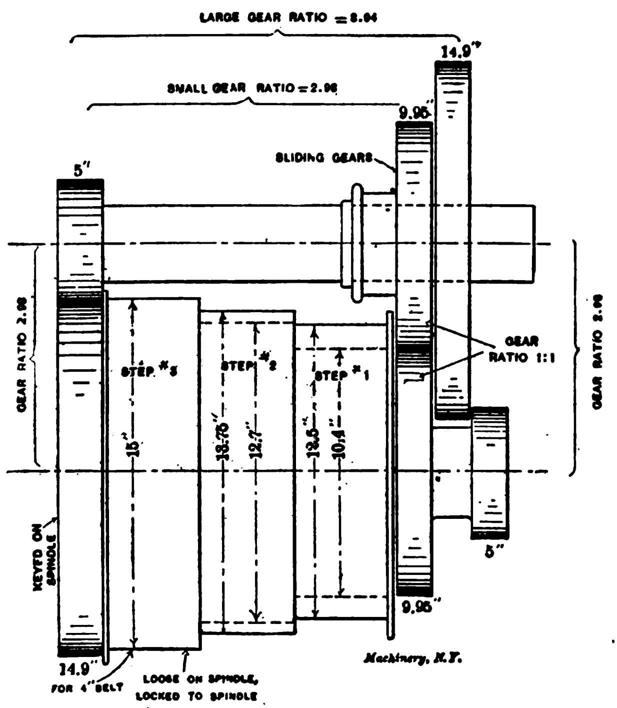


Fig. 2. Two Methods of Laying out the Cone for a Double Back-Geared Spindle

any collection of machine tools, even in those of modern design. It is possible to design a cone so as to obtain speeds in strict accordance with the geometrical series. In most cases the counter-shaft cone and the one on the machine are made from the same pattern, so that it is necessary that the diameters be the same for both cones, and since the belt is shifted from one step to another, its length must be kept

constant. This is accomplished by having the sum of diameters of corresponding steps equal.

We will take as the large diameter of the cone, 15 inches. The ratio of the speeds off step 1 and step 3 is 1.44 to 1. This ratio also equals $\frac{D \times D}{d \times d}$ where D is the diameter of largest step and d is the diameter of smallest step. Making them opposite terms in an equation we get,

$$1.44 = \frac{D \times D}{d \times d} = \frac{D^2}{d^2}$$

$$\text{or } 1.44 \times d^2 = D^2$$

$$d = \sqrt{\frac{D^2}{1.44}} = \sqrt{\frac{15 \times 15}{1.44}} = 12.5 \text{ inches, diameter of small step.}$$

The sum of the corresponding diameters on the cones is $15 + 12.5 = 27.5$.

Since this is a three-step cone the middle steps must be equal. Therefore $\frac{27.5}{2} = 13.75 =$ diameter of middle step. We found that the ratio of the speeds off first and second step is 1.2. Let us examine the above figures to see that the diameter of the middle step is correct. Thus,

$$\frac{15}{12.5} \times \frac{13.75}{13.75} = 1.2,$$

which is the correct ratio. This cone is shown in full lines in Fig. 2.

Let us now figure the diameter of the back-gears. We will assume that the smallest diameter possible for the small gears in the set is 5 inches. In order to keep the gears down as small as possible we will take this figure as the diameter of the small gear here. It is general practice, though obviously not compulsory, to make the two trains in a set of back gears equal as to ratio and diameters. When double back gears are used, the large ratio set is made with two trains of similar ratio. The small ratio set is then composed of two trains of gears whose ratios are unlike. The ratio of each train in the large ratic set, if taken as similar, is equal to the square root of the whole ratio; thus, in our drive we have $\sqrt{8.94} = 2.98$, and from this the large gear is $5 \times 2.98 = 14.9$ inches in diameter. The ratio of the small ratio set is equal to 2.98, and as one train of gears in the double back gear arrangement is common to both sets, the remaining train in the small ratio set must be of equal diameters, or $5 + 14.9 \div 2 = 9.95$ inches, as shown in Fig. 2. These figures will have to be slightly altered in order to adapt them to a standard pitch for the teeth, which part of the subject we will not deal with here.

In order to be able to compare the results of the two different methods of selecting counter-shaft speeds mentioned above, let us figure out the dimensions of a drive with counter-shaft speeds arranged according to the second method.

Proceeding in a manner similar to that pursued for the case treated above, we may tabulate the speeds as shown in the table on next page.

CONE.	Open Belt.		Small Ratio Gears in.		Large Ratio Gears in.	
	Fast Counter Speed.	Slow Counter Speed.	Fast Counter Speed.	Slow Counter Speed.	Fast Counter Speed.	Slow Counter Speed.
Step 1.....	223	186.	74.4	62.	24.85	20.7
Step 2.....	155	129.	51.6	43.	17 25	14.4
Step 3.	107	89.4	35.8	29.8	12.	10.
	1	2	3	4	5	6

The various ratios are:

Large ratio gears.. 8.94 to 1
Small ratio gears.. 2.98 to 1
Counter-shaft speeds .... 1.2 to 1
Speeds off step 1 to those off step 2.................. 1.44 to 1
Speeds off step 1 to those off step 3.................. 2.07 to 1

The cone dimensions are figured in the same manner as before and are 10.4 inches for step 1; 12.7 for step 2; 15 for step 3. This cone is shown dotted in Fig. 2.

We are now in a position to compare the results given by the two methods above referred to. Let us make the first comparison from the point of view of power delivered by the belt. It is well-known that the power of a belt is directly proportional to the speed at which it runs. This fact gives us an easy means of comparing our two designs. We will do this by charting the speed in feet per minute of the belt when running on the different steps of the two cones for each spindle speed. This has been done in Fig. 3, where the full lines show the curve for the first method, and the dotted lines show that for the second method. The curves at the left are those for the slow counter speeds, while at the right are seen those for the fast counter speeds. Attention is called to the great difference in power delivered between the two counter speeds in the first case, while the two sets of curves for the second method lie close together. Also, note the gain in power at speeds obtained through the slow counter in the second case. The power lost in the second case on the fast counter speeds will not be felt so much, for the same principle applies here as it does to the strength of beams, bridges, etc., *viz.*, a chain is no stronger than its weakest link.

The constant-speed pulley drive has become quite a common feature in machine tool design, and has become quite a strong favorite with many. Had our machine been provided with a drive of this design, we would have had a curve on the chart as shown by the vertical full line. The power delivered by the belt would have been constant throughout the full range of speed. This curve also applies to the motor drive, when a constant-speed motor, or a variable-speed motor of the field control type, is used, although slight modifications would have to be made for the decrease in efficiency at the extremes of the

speed range of the latter type motor, which would cause a slight bend in the curve, making it convex toward the right. Motors using the multiple-voltage system, or the obsolete armature resistance control, would show curves quite as irregular as those from the cone and back-gear drive.

Another method of comparison is by charting the pull or torque at the spindle for each spindle speed. This is done in Fig. 4, where the

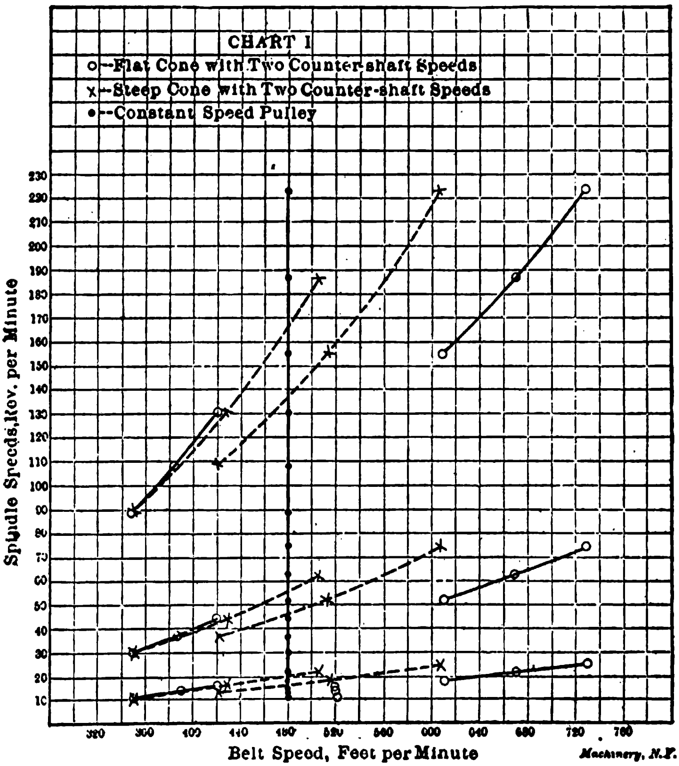


Fig. 3. Variation in Belt Speeds for Various Methods of Driving

constant speed pulley drive is shown by the full line, and is used as a comparator by which to compare the results of the two drives treated above. This figure is self-explanatory and will not need to be interpreted, but attention may be called to how much better the drive of the second case follows the ideal line than does that of the first method. This chart also shows how very close a cone and double back-gear drive comes to the constant belt-speed drive with equal power at all speeds.

Much has been said about the relative values of the two styles of cone pulleys treated above, but the charts given herewith will no doubt surprise some, and may be the means of turning them in favor

of the second method. The only good point the first method has over the second is in the appearance of the cone which has, apparently, powerful lines, which are, however, misleading, as has been shown.

Another disadvantage of the first method is the wide ratio of the

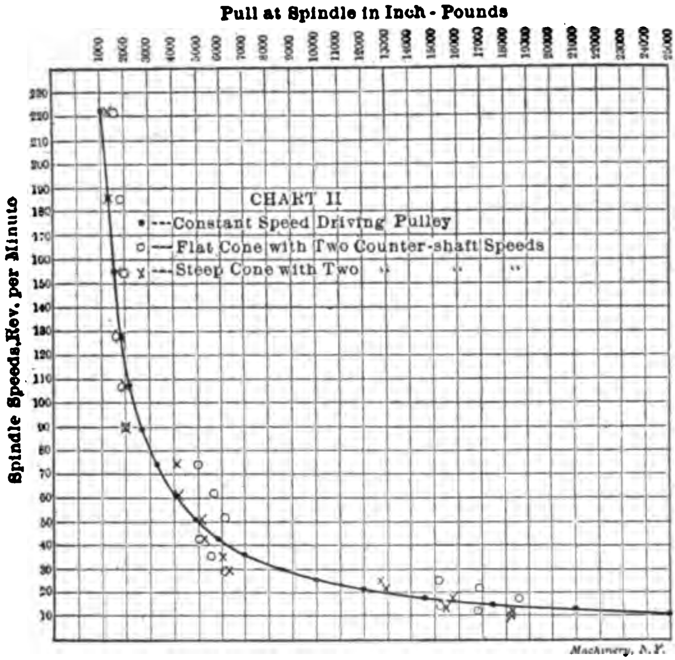


Fig. 4. Comparison of Torques for Various Methods of Driving

counter-shaft speeds, where, in order to get sufficient power out of the slow speed counter-shaft belt, we must have the high-speed pulley running at almost prohibitive speed, which soon tells, and as loose pulleys are a source of annoyance when their speed is moderate, trouble is sure to appear when the limit of speed is approached.

CHAPTER IV

GEARED OR SINGLE PULLEY DRIVES

Whether the geared drive, so called in order to distinguish it from the belt drive used with stepped cone pulleys, originated with some machine tool builder who was desirous of improving a given machine, or whether it was first suggested by a machine tool user in an endeavor to secure better facilities for machine operation, would be interesting to know, but difficult to determine.

Whatever the origin, the geared drive is a response to a demand for a better method of speed variation than could be obtained from stepped pulleys and a movable belt. The gradually growing demand for more powerful machine drives in the past has led to the widening of belts to the maximum point consistent with a desirable number of steps of the pulley, and the ease of belt shifting. The limiting point for belt width may be said to be reached when a belt can no longer be shifted easily by hand. For some machines, notably lathes, the maximum diameters of the driving pulleys are generally limited by conditions inherent in the machine themselves.

Back-gears were in many instances increased in ratio to make up for what could not be had by further increase of belt widths or pulley diameters, until in some cases the gap between speeds obtained directly by the belt and those obtained through the back gears became too great. When such conditions were reached, obviously, the next suggestion involved the combination of a constant speed belt of such a width and operated at such a speed as to give the requisite power, in connection with some combination of gears to be used for obtaining the desired variation in speeds. Such a combination is, in fact, a reversion of type; a going back to a system of driving formerly much used by foreign builders of machine tools. Many foreign builders objected to the use of stepped pulleys, considering their use as a deviation from, or, as being contrary to, good mechanical practice, preferring in many cases to secure speed variation by means of separate changeable gears. The objectionable feature of such a system did not suit American ideas, hence the early adoption of stepped pulleys and a movable belt as a means of quickly effecting changes even though the device was and is still considered by some designers as anomalous or paradoxical from the standpoint of pure mechanics. The substitution of the variable speed geared drive for the stepped pulley drive is therefore not due to any inherent defect in the stepped pulley so much as to its limitations as previously mentioned, and to a desire for improved facilities for quickly obtaining speed variations.

For belt-driven machines that require a variable speed, the geared drive will probably come more into use whenever its adoption will be justified from a productive or a commercial standpoint. Whatever

defects may be existent in any of its varied forms will be tolerated just as long as it meets and fulfills required conditions.

As a device of utility the geared drive has passed the point where it might by some have been considered as a fad. As a matter of fact, scarcely any new device representing a radical departure from generally accepted design and practice has ever been brought out that was not considered a fad by some one. The history of machine tool progress has shown that the fad of yesterday has frequently become the custom or necessity of to-day. Extreme conservatism will see a fad where progress views an undeveloped success. One drawback to the general adoption of any geared drive is its cost, and this will determine in most cases whether it or a belt drive shall be used; it is a matter requiring careful judgment to determine the point where the results obtained justify the added expense.

It is, however, with very few exceptions, the opinion among builders and users of machine tools that the single pulley drive will largely supersede the cone drive. Still for certain conditions it is doubtful whether we will find anything better than our old servant, the cone. The two principal advantages possessed by the single pulley drive are: First, a great increase in the power that can be delivered to the cutting tool owing to the high initial belt speed. The belt speed always being constant, the power is practically the same when running on high or low speeds. The cone acts inversely in this respect; that is, as the diameter of the work increases, for a given cutting speed, the power decreases. As a second advantage, the speed changes being made with levers, any speed can be quickly obtained.

To these might be added several other advantages. The tool can be belted direct from the lineshaft; no counter-shaft is required; floor space can be economized. It gives longer life to the driving belt; cone belts are comparatively short-lived, especially when working to their full capacity. There are, however, some disadvantages to be encountered. Any device of this nature, where all the speed changes are obtained through gears, is bound to be more or less complicated. The first cost, as mentioned, is greater. There is also more waste of power through friction losses. A geared drive requires more attention, break-downs are liable to occur, and for some classes of work it cannot furnish the smooth drive obtained with the cone. Most of these objections, however, should be offset by the increased production obtained.

To the designer the problem presented is one of obtaining an ideal variable speed device, something that mechanics have been seeking for years with but poor success, and it is doubtful whether we will get anything as good for this purpose as the variable speed motor in combination with double friction back-gears and a friction head. There are, it is true, some very creditable all-gear drives on the market in which the problem has been attacked in various ways. Still there is ample room for something better. The ideal single pulley drive should embody the following conditions.

1. There should be sufficient speed changes to divide the total range

into increments of say between 10 and 15 per cent.

2. The entire range of speeds should be obtained without stopping the machine.

3. Any speed desired should be obtained without making all the intermediate changes between the present and desired speed.

4. All the speeds should be obtained within the tool itself, and no auxiliary counter-shaft or speed variators should be used.

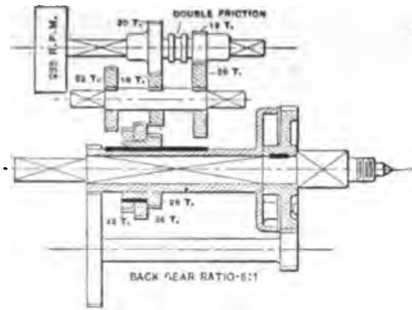
5. Only the gears through which the speed is actually being obtained should be engaged at one time.

6. The least possible number of shafts, gears and levers should be used.

There are few subjects in machine design which admit of so many combinations, arrangements and devices. In Figs. 5 to 10, inclusive, are shown some examples taken at random from a large collection. All of these, except Fig. 10, have the number of teeth and the speeds marked. Each has some good points, but none of them possesses all the points referred to above. The only reason for showing them is to show what a vast number of designs can be devised. One of them, that shown in Fig. 5, has been built, a number of machines have been running for over a year, and they give very good results. In Fig. 11 is shown the way the idea was worked out, as applied to a 20-inch Le Blond lathe.

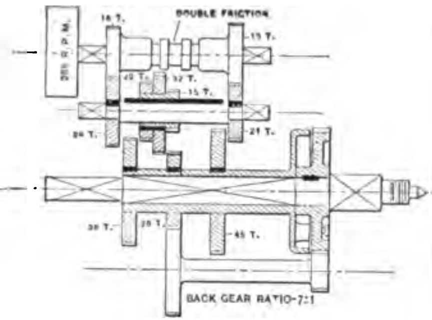
The design for the headstock shown in Fig. 11 needs little explanation since the drawing shows the parts quite clearly. The friction clutch on the driving-shaft *Z*, which alternately engages pinions *H* and *J*, is of the familiar type used in the Le Blond double back-geared milling machine. Sliding collar *D*, operated by handle *S*, moves the double tapered key *E* either to the right or left as may be desired, raising either wedge *W* or *W'*, which in turn expand rings *X* or *Y* within the recess in either of the two cups, *F* and *F'*. Either of two rates of speed is thus given to quill gear *K* and the two gears *L* and *M* keyed to it. On the spindle is a triple sliding gear which may be moved to engage *P* with *M*, *O* with *L* (as shown in the drawing) or *N* with *K*, thus giving three changes of speed when operated by lever *T*. The six speeds obtained by the manipulation of levers *S* and *T* are doubled by throwing in the back-gears, giving 12 speeds in all.

In comparing the merits of a series of gear drive arrangements like those shown in Figs. 5 to 10, one might apply the "point" system in determining the most suitable one. The number of points that are to be assigned to a device for perfectly fulfilling any one of the various requirements would be a matter requiring nice discrimination. So the method outlined below is to be taken as being suggestive, rather than authoritative. The first requirement is that there shall be sufficient speed changes to divide the total range into increments of between 10 and 15 per cent. The six schemes proposed do not all, unfortunately for our proposal, take in the same range of speed; considering, however, that they were each to be designed to give from 9 to 240 revolutions per minute to the spindle, as in case Fig. 5, and that a 15 per cent increment is to be allowed, the number of changes required



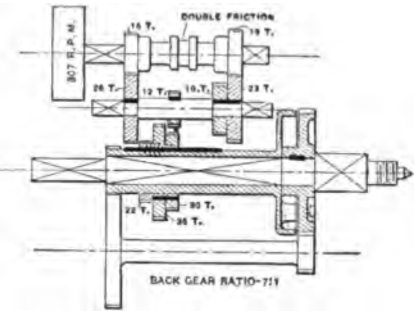
SPINDLE SPEEDS
B. GEARS IN-9.-12.4-16.7-21.6-29.78-40
B. GEARS OUT-54.-74.-100.-129-179-240

Fig. 5



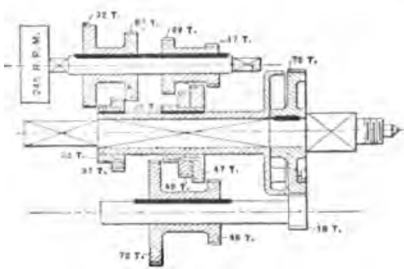
SPINDLE SPEEDS
9-12-16-24-32-48
63-84-112-168-224-336

Fig. 6



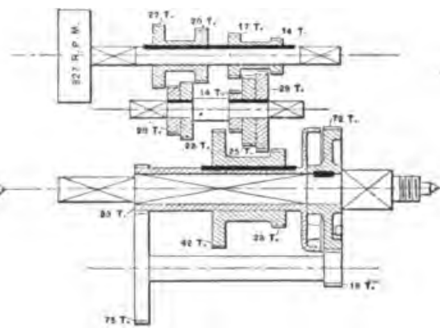
SPINDLE SPEEDS
9-12-16-24-32-48
63-84-112-168-224-336

Fig. 7



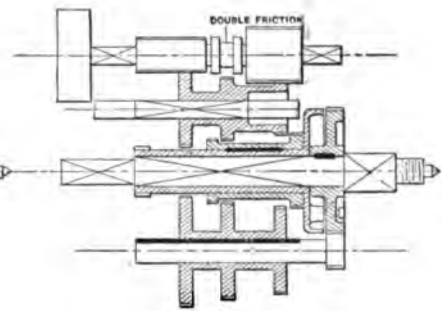
SPINDLE SPEEDS
T. GEARS OUT-6-8-11-16
1ST B. GEARS IN- 22-32-45-62
2ND B. GEARS IN-89-128-179-248

Fig. 8



SPINDLE SPEEDS
B. GEARS IN-6-8-11-14-19-26-33 40
B. GEARS OUT-54-72-95-136-164-222-297-360

Fig. 9



SPINDLE SPEEDS
24 SPEEDS OBTAINABLE
NO DATA GIVEN

Fig. 10

can be found in the usual way by dividing the logarithm of 27— the total speed ratio required (240 ÷ 9 = 27)—by the logarithm of 1.15, which is the ratio of the geometric series desired. This gives 24 speeds, about, as needed to meet the requirements. Suppose we assign 15 points to a machine having 24 speeds. Let us set this down in its proper place in the suggested table given below. For the second qualification, that the machine shall not have to be stopped, we may assign 20 points to the ideal machine. The principle of "selective" control is assigned 10 points. The fourth consideration, requiring that all speeds shall be obtained within the tool itself is a positive requirement. If it is not met, the mechanism is out of the contest, so this question need not be considered in our table of points. Fifteen points are suggested for the requirement that the gears not in use shall not be running in mesh. The sixth requirement reads "The least possible number of shafts, gears and levers should be used." It is suggested

A SUGGESTED TABULATION OF THE MERITS OF THE VARIOUS DRIVES PROPOSED

Requirements.	Perfect Design.	No. 1	No. 2	No. 3	No. 4	No. 5	No. 6
No. of changes required compared with No. obtained..	15	8	8	8	8	10	15
Stopping of machine	20	14	14	14	14	14	14
"Selective" control........	10	7	7	7	7	7	7
Gears not in use, must not be in mesh	15	13	13	13	15	15	13
Ratio of No. of changes to No. of movements	20	15	15	15	13	12	14
Ratio of No. of changes to No. of gears............	20	10	9	9	9	16	18
Total	100	67	66	66	66	74	81

that this be divided, giving 20 points to the question of the ratio of the number of changes obtained to the number of movements required of the operator to obtain them, and giving the same number of points to express the ratio of the number of changes obtained to the number of gears used in obtaining them. The sum of these points added together is 100, which may be considered as representing the ideal design.

In filling out the table, since Fig. 5 has only 12 speeds or half the number required, we will give it only one-half the number of points, dealing similarly with the other designs up to No. 6, in Fig. 10, which is perfect in this respect. The machine has to be stopped to throw in back-gears. Assuming that this would not have to be done in 70 per cent of the changes, we get a uniform value of 14 for this consideration for all the cases. The feature of selective control is only about two-thirds realized in any of these designs, since the triple sliding gear used in all of them, in moving from one extreme to the other, passes through an intermediate position which is not required at the time.

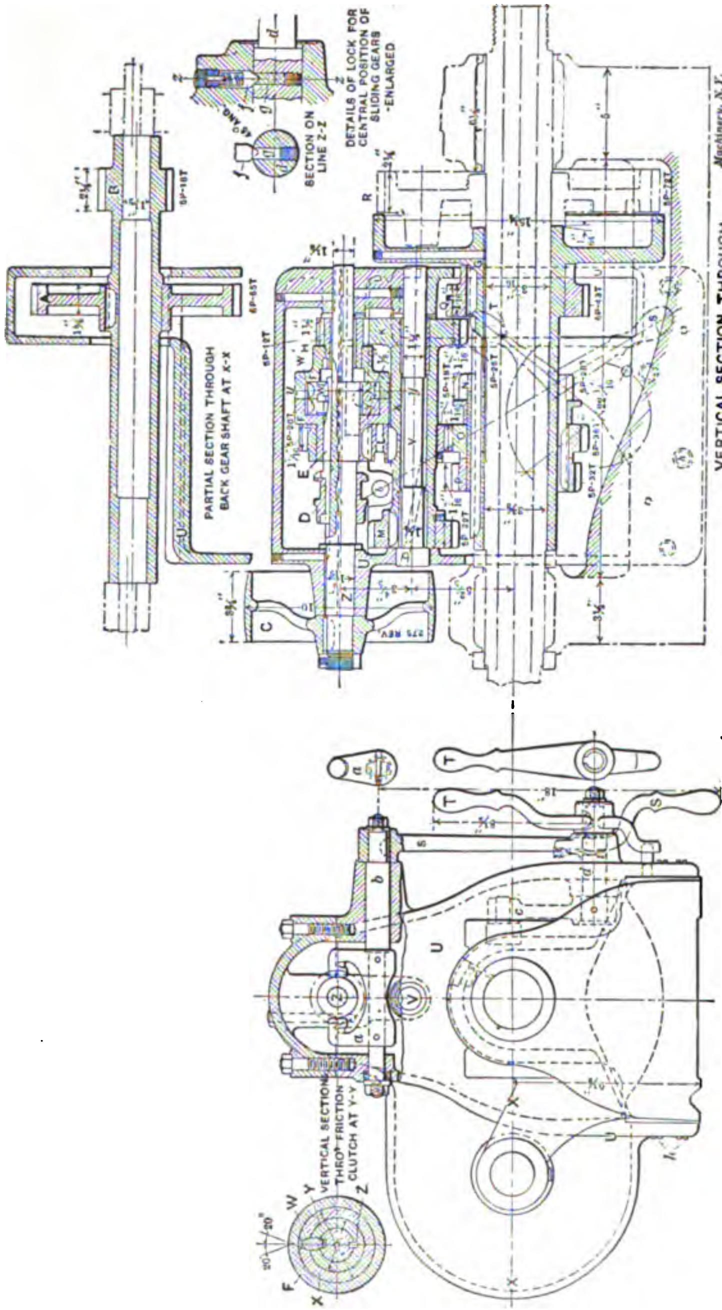


Fig. 11. The Layout shown in Fig. 5 applied to Head-stock of a 20-inch Lathe

We may therefore assign the value 7 to each of these designs on this account. As to the question whether the gears not in use are running idly in mesh, all the designs are nearly perfect. The values set down in this table are suggested by this consideration. In considering the number of movements required to effect the number of changes obtained, the throwing in of the back-gear is credited with four motions, the stopping of the machine, unlocking of the spindle from the gear, the throwing in of the back-gears, and the starting of the machine. The 20 points of the ideal machine are then multiplied by each of the ratios obtained by dividing the number of changes by the number of movements, and the number of points found are set down as shown. For the last item, twice as many changes as there are gears employed is taken as a maximum which can probably not be exceeded. With this as a standard, the ratio obtained by dividing the number of changes by the number of gears used is employed to calculate the number of points. Adding the number of points obtained in each column we find that No. 1 has 67, No. 2, 3, and 4 each have 66, while No. 5 has 74, and No. 6, 81.

The comparison has been undertaken in this way with the understanding that all the arrangements are susceptible of being embodied in a practicable design. That arrangement No. 6 is practicable is strongly to be doubted. The number of teeth in the various gears used are not given, and it is far from probable that one could obtain with this arrangement a series of speeds in geometrical progression by moving in regular order the three levers required. Nos. 4 and 5, while otherwise well arranged, are open to the objection that sliding gears rotating at high rates of speed are used. This, if valid, constituted a disqualifying objection similar to that mentioned in relation to the fourth requirement. The first three cases in which a friction clutch instead of sliding gears is used on the driving shaft are therefore much to be preferred for this reason. Of these first three cases, our tabulation shows that case No. 1 has a slight advantage, and Fig. 11, in which this arrangement has been applied to a 20-inch lathe headstock, shows that the scheme is a simple and satisfactory one, so far, at least, as one can judge from a drawing.

CHAPTER V

DRIVES FOR HIGH-SPEED CUTTING TOOLS

What has been considered in the past as marvelous in the performance of high-duty cutting tools may now be compared with the proved results of air-hardening cutting tools. The metallurgist has proved to us, and a great many machine tool builders have satisfied themselves by practical experiment, that the high-speed cutting steels are at our service, but they must be properly shod if they are to be used to the best advantage. Some concerns who have experimented with the high-speed steels, and who anticipated much, have failed through lack of a proper analysis of the conditions which accompany the use of the high-speed cutting steels. It takes but a moment's reflection to convince one of the absurdity of trying to get as effective a fire from a six-inch as from a thirteen-inch gun, even though the same explosive charge is used in both.

Some viewed this unusual commotion about the high-speed cutting steels as being somewhat fanatical or a fad which would rage for a time, and then die a natural death, as many others have done. True, this was not the first high-duty cutting steel which had been advanced with enormous claims of efficiency. Mushet steel had been on the market for several years, and the great things predicted for it did not fully meet everybody's expectations. The chief reason for this was its far too limited use in a great many cases, on account of its being expensive, difficult to forge, grind, and to get a satisfactorily finished surface with it, and the failure of the machine to stand up to the chip it could take. Then again, when Mushet steel was introduced, competition among machine tool builders for increased product from their machines did not begin to compare with that which now exists with firms which more than ever are on an intensely manufacturing basis. Manufacturing plants of any considerable size using metal cutting tools are bidding nowadays for special machinery of the simplest form to augment the output of a single product, and not comparatively complicated combination tools, designed for many operations on many pieces, and which save considerable room and first cost of installation, but are of necessity inconvenient, and unsuitable for high-duty service.

The complaint which has been made by some that the new high-speed cutting steels are unfit for finishing surfaces cannot be consistently sustained. The modernly-designed manufacturing grinder has unquestionably proved to be the proper tool for finishing surfaces from the rough; and undoubtedly, and beyond peradventure, the grinder is the natural running mate for the high-duty turning lathe and planer; and it seems probable that, instead of the grinder being a rarity and a luxury in shops, as a sort of tool-room machine, it

will be as much in evidence for manufacturing purposes as the more commonly-known machine tools of the present, or more so.

The innovations of the day in machine tool evolution are in most remarkable harmony and synchronism. The electric motor, which is fast developing the independent machine drive, demands a high speed for maximum efficiency of the motor; and what do we find contemporaneously developed but the high-speed cutting steels, the practicable commercial grinder, and the comparatively high-speed non-stroke milling machine to supersede the comparatively slow multi-stroke planer? Unquestionably, there never has been in the whole history of the machine tool business such an opportunity for the enterprising capitalist, the engineer, and the designer, to invest their money, brains and skill in a type of machine tools that will be as different from the present type of machine tools as the nineteenth century lathe is from the simple and crude Egyptian lathe of tradition.

The development of the cutting or producing end of the machine appears to be further advanced than the driving end. The direct motor drive without inter-connecting belts, chains, and gears is undoubtedly the simplest, most convenient, and most effective. The motor which is most desired has not been designed, but it should be a comparatively slow-speed motor having high efficiency, whose speeds vary by infinitesimal steps between its minimum and maximum limits, fully as simple as the "commutatorless" type, and with far higher pressures than are now used. In the meantime, during the process of development, we shall have to be content with the usual compounding elements between the motor and the driving spindle; but these compounding elements, in order to keep up with the procession, will naturally undergo revolutionary changes in design.

The silent chain drive and the high-speed motor are mutual helpmates; geared variable speed devices and single-speed induction motors are well wedded, but cone pulleys are practically just beginning to receive that examination and attention which can fit them for the service of higher speeds.

In the case of a turning lathe, as would naturally be expected, we are very much limited in the range of the sizes of pieces that can be turned—if we maintain an efficient range of speeds and sufficient diameters and widths of pulleys for surface speeds of belts—unless we use an abnormally ponderous cone pulley, which is entirely out of the question. To make this point clear, it may be well to analyze a specific case. We will assume that the lathe is designed with a four-stepped cone and with "front-gears" (the speed ratios of front-gears are figured the same as back-gears, but their thrust at the front box is opposite in direction to that of the back-gears and to the lifting effect of the tool, as it properly should be), two countershaft speeds, and for cutting 30-point carbon steel at a speed of 100 feet per minute with a chip of 5/16 by 3/32 inch cross section. It is furthermore assumed that the work and cutting tool are rigidly supported, and that the cutting tool has the proper amount of rake for least resistance and a fair amount of endurance.

Calculation of Cutting Force of Tool, and Speed of Belt

In order to make absolute computations of the required diameters, we should have reliable data on the amount of cutting force at the cutting edge of the tool when cutting the various metals at high speeds, reliable data for the best efficiency of the redesigned machine, and the approximate distance between the centers of the driving spindle and counter-shaft. Several experiments were made by Hartig, and subsequently by others, on the horse-power required at the cutting edge of a tool when cutting various metals at slow speeds with the ordinary tempered steels. The horse-power was determined by multiplying the weight of chips turned off per hour by a constant whose value varied with the degree of hardness of the metal cut and the conditions of the cutting edge of the tool. The average of the several constants for about 30-point carbon steel seems to be about 0.035.

Hartig's expression is given in the formula

$$H.P. = cW = 0.035 \times \pi \times D \times n \times d \times t \times 0.28 \times 60 \qquad (9)$$

and the usual expression for horse power is given in the form,

$$H.P = \frac{FS}{33000} = \frac{F \times \pi \times D \times n}{33000 \times 12} \qquad (10)$$

in which

$H.P.$ = horse power absorbed at the cutting edge of tool.
c = constant 0.035.
W = weight of chips per hour.
D = mean diameter of the area turned off per hour.
n = revolutions per minute.
d = depth of chip.
t = thickness of chip.
0.28 = assumed average weight per cubic inch of 30-point carbon steel.
F = force at cutting edge of tool.
S = distance through which force F acts.

Equating (9) and (10),

$$F = 0.035 \times 0.28 \times 60 \times 33000 \times 12 \times d \times t = 232850\ dt.$$

Since the chip assumed to be cut is 5/16 by 3/32 inch cross section, then the force at the cutting tool is

$$F = 232850 \times 5/16 \times 3/32 \text{ inch} = 6820 \text{ pounds.}$$

If the cutting speed is 100 feet per minute then the work at the tool

$$W = 6820 \times 100 = 682000 \text{ foot-pounds.}$$

If the efficiency of the machine is assumed at 85 per cent, then the effective work of the belt must be

$$W = \frac{682000 \times 100}{85} = 802500 \text{ foot-pounds.}$$

We will assume that a 5-inch double belt is the practical limit for the belt which can be conveniently used on the machine, and that the effective pull is 70 pounds per inch width when wrapped around a cast-

iron pulley with a contact surface of 180 degrees. The total effective pull is then

$$5 \times 70 = 350 \text{ pounds.}$$

Since our belt must deliver 802500 foot-pounds per minute, its velocity will be

$$V = \frac{802500}{350} = 2295 \text{ feet per minute, approximately,}$$

which must be proportional to the diameters of the cone pulleys and the counter-shaft speeds, which are obtained as follows.

It is customary to consider speeds in a series of geometrical progression if the most efficient and convenient range of speeds is desired. The constant multiplier will then be

$$r = \left(\frac{l}{a}\right)^{\frac{1}{n_1 - 1}}, \text{ in which} \tag{11}$$

r = constant multiplier.
l = maximum R. P. M. of spindle.
a = minimum R. P. M. of spindle.
n_1 = number of speeds.

Let it be assumed that the lathe is designed to turn sizes from 1 to 6 inches. The corresponding maximum and minimum revolutions per minute for the cutting speed 100 feet per minute are 382 and 62, approximately. Then from (11)

$$r = \left(\frac{382}{62}\right)^{\frac{1}{15}}$$

$$\log r = \frac{1}{15} \log 6.16$$

$$r = 1.128$$

The whole series of speeds in geometrical progression and the diameters of stock, which will approximately correspond, if a cutting speed of 100 feet per minute be used, is given in the following table:

SPEEDS IN R.P.M.	DIAMETER OF STOCK			SPEEDS IN R.P.M.	DIAMETER OF STOCK		
382	1	CONE PULLEY SPEEDS	FAST COUNTERSHAFT SPEED	145	$2\frac{5}{8}$	CONE PULLEY SPEEDS	SLOW COUNTERSHAFT SPEED
338	$1\frac{1}{8}$			128	3		
300	$1\frac{1}{4}$			113	$3\frac{3}{8}$		
265	$1\frac{1}{2}$			101	$3\frac{3}{4}$		
235	$1\frac{5}{8}$	BACK GEAR SPEEDS		89	$4\frac{5}{16}$	BACK GEAR SPEEDS	
208	$1\frac{13}{16}$			78	$4\frac{7}{8}$		
184	$2\frac{1}{16}$			69	$5\frac{1}{2}$		
163	$2\frac{3}{8}$			62	6		

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In Fig. 12, assume that the counter-shaft and spindle cone pulleys are the same size, as is usually the case for the engine lathe. Let

D_1 = diameter of largest step.
D_4 = diameter of smallest step.
n' = slowest speed of countershaft.

N_1 = fastest speed of spindle to correspond with slowest countershaft speed.

N_4 = slowest speed of spindle without back-gears to correspond with slowest countershaft speed.

$$\text{Let } \frac{D_1}{D_4} = r \tag{12}$$

$$\frac{D_4}{D_1} = \frac{1}{r} \tag{13}$$

Then

$$n' \times r = N_1 \tag{14}$$

$$n' \times \frac{1}{r} = N_4 \tag{15}$$

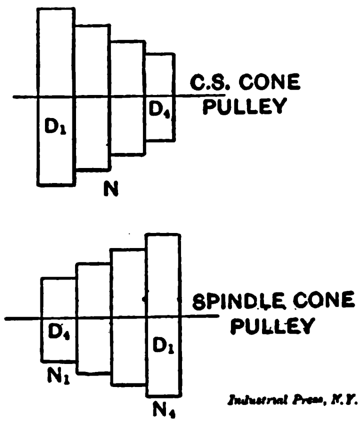

Fig. 12

Combining (14) and (15),

$$n' = \sqrt{N_1 \times N_4} \tag{16}$$

Substituting in (16) the proper speeds taken from the table;

$$n' = \sqrt{145 \times 101} = 121$$

From (14)

$$r = \frac{N_1}{n'} = \frac{145}{121} = 1.199$$

$$D_4 = \frac{V}{\pi n'} \tag{17}$$

Substituting in (17) the value of V and n',

$$D_4 = \frac{2295 \times 12}{3.14 \times 121} = 72\tfrac{1}{2} \text{ inches.}$$

From (12)

$$D_1 = r \times D_4 \tag{18}$$

Substituting in (18) the value of r and D_4

$$D_1 = 1.199 \times 72\tfrac{1}{2} = 87 \text{ inches.}$$

The front gear ratio from spindle cone speed to driving spindle speed will be $\frac{145}{89} = 1.629$.

Since the values of the constants used in computing the force at the cutting tool were taken from experiments made with slow cutting speeds, and would be considered low in view of the fact, noted by some, that the work at the tool for high speeds increases in far greater proportion than the increased cutting speeds; and since the assumed 70 pounds per inch width for effective pull at the belt is quite liberal, it is clear that the pulleys are practically at a minimum size under the conditions assumed. It is therefore convincingly apparent that for the ordinary back-geared head, belts can be of no avail for high-speed cutting except for extremely limited ranges of diameters of stock.

If the diameters of the pulleys are reduced by speeding up the belts and gearing down the spindle, nothing is availed in most cases but an added and useless expense, since every compounding element is a loan for a mortgage whose interest rates sometimes increase pretty nearly in a geometrical progression.

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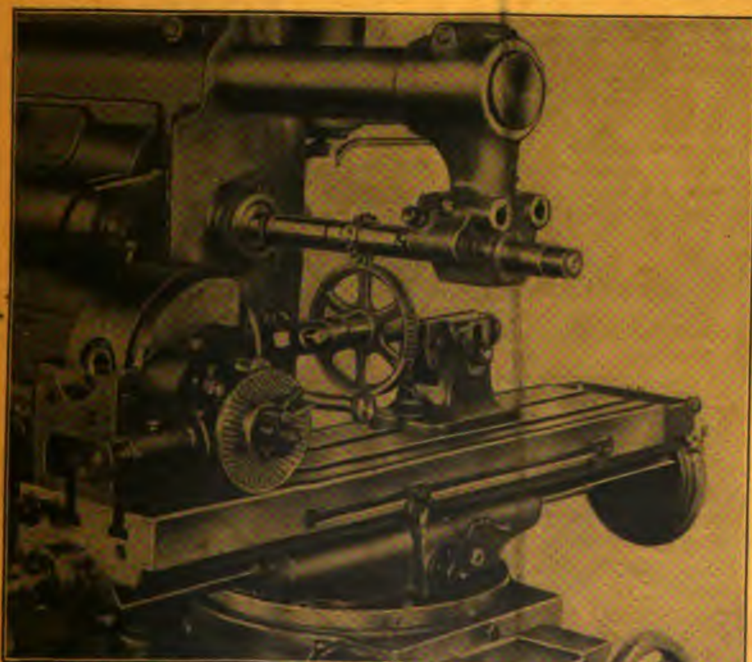

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CONTENTS

When preparing the third edition of this Reference Series book, a considerable amount of matter pertaining directly to general shop calculations was added. In order to provide space for this material, the chapters on more advanced shop calculations, including Square and Square Root, Use of Formulas, and Use of Tables of Sines, Cosines, Tangents and Cotangents, were omitted, and are, together with additional matter of an advanced nature, included in MACHINERY'S Reference Series No. 52, Advanced Shop Arithmetic for the Machinist.

INTRODUCTION

In the following, some of the most common shop subjects requiring simple calculations have been treated, and special efforts have been made to treat each subject as simply as possible, so that the present treatise may be of service to those, particularly, who have not previously acquired a great amount of knowledge about handling figures, and who are not familiar with mathematical expressions and usages. In order to fix the processes and rules more firmly in the reader's mind, examples have been given in almost all instances, and in many cases a number of similar examples have been given, so as to permit the repetition of the same calculation a number of times. Practically all formulas have been written out in words, as this gives a better idea about what the formula actually means, at least to those not familiar with engineering handbooks. Mathematical signs have also been avoided to a certain extent, and the corresponding words have been written out in full. In short, all precautions have been taken to present the methods in as plain and simple language as possible. Many text-books deal with principles rather than with specific examples, and to a person who is not used to solving problems of the kind that are met with in the machine shop, it is often difficult to apply the principles involved to each particular case. The purpose of this book has, therefore, been to select the most common specific cases, and to show directly how the principles are applied.

While the subject in hand has been treated to accommodate the requirements of those who demand a book that is plain and simple, it has been necessary to presuppose fundamental knowledge in regard to the use of numbers in calculations, that is, the reader must be fairly competent to add, subtract, multiply, and divide whole numbers and decimals, and also have some fundamental ideas of the use of common fractions.* If such knowledge has been acquired, no difficulty will be experienced in making use of the rules and formulas given.

It is assumed that the reader is familiar with the common mathematical signs, $+$ (plus) which signifies addition, $-$ (minus) which signifies subtraction, $\times$ (times) which signifies multiplication, and $\div$ (divided by) which signifies division, as well as with the sign $=$ (equals) which is put between quantities which are equal to one another to signify this condition. But it may be appropriate to call attention to the different methods commonly used for indicating division, as these may not be clear to all. Usually, as we already have said, in arithmetic, division is indicated by the sign $\div$, so that we have, for instance,

$$12 \div 3 = 4.$$

A more common method in engineering books, however, is to simply

* For a simple treatise on these elements, see MACHINERY'S Jig Sheets Nos. 1A to 15A, inclusive.

write the dividend as the numerator of a fraction and the divisor as the denominator, thus:

$$\frac{12}{3} = 4.$$

In that case the fraction indicates a division. This system will be followed in many of the following formulas, and it should therefore be remembered that *the line between the numerator and denominator in a fraction always indicates a division, the numerator to be divided by the denominator.*

The actual division, however, is not necessarily worked out in every case where division is thus implied. When two divisions are multiplied together, cancellation and the following operations of addition or subtraction may make the actual numerical work very simple.

Although knowledge of common fractions is presupposed, it may be well at this point to repeat the rules for multiplication and division of common fractions, as in the following many operations of this kind must be made. Two fractions are multiplied by multiplying numerator by numerator and denominator by denominator, (*numerator* being the *upper*, and *denominator* the *lower* quantity in a fraction). For instance, let it be required to multiply ¼ by ⅜. We have then,

$$\frac{1}{4} \times \frac{3}{8} = \frac{1 \times 3}{4 \times 8} = \frac{3}{32}$$

If the numbers to be multiplied contain whole numbers, these are first converted into fractions. Let it be required to multiply 1¼ by 3¼. We have then,

$$1\frac{1}{4} \times 3\frac{1}{4} = \frac{5}{4} \times \frac{13}{4} = \frac{65}{16} = 4\frac{1}{16}$$

Division is simply the reverse of multiplication. The number which is to be divided is called the *dividend*, and the number by which we divide is called the *divisor*. If one number is to be divided by another, we simply invert the divisor, and *proceed as in multiplication.* To invert the divisor means that we place the denominator as numerator, and the numerator as denominator; for instance, 3/8, inverted, is 8/3. Suppose that we wish to divide 3/4 by 7/16. We have then,

$$\frac{3}{4} \div \frac{7}{16} = \frac{3}{4} \times \frac{16}{7} = \frac{48}{28} = 1\frac{20}{28} = 1\frac{5}{7}$$

If the number to be divided contains a whole number besides a fraction, we first convert this into a fraction, and then proceed as before. Suppose that we wish to divide 2¼ by 3¾. We have then,

$$2\frac{1}{4} \div 3\frac{3}{4} = \frac{9}{4} \div \frac{15}{4} = \frac{9}{4} \times \frac{4}{15} = \frac{36}{60} = \frac{3}{5}$$

A parenthesis about a mathematical expression indicates that the calculation enclosed by the parenthesis is to be carried out before the other calculations in the example; thus, $(5 + 3) \times 2 = 13 \times 2 = 26$, but $5 + (8 \times 2) = 5 + 16 = 21$.

CHAPTER I

FIGURING TAPERS

In all circular or round pieces of work, the expressions "taper per inch" and "taper per foot" mean the taper on the *diameter*, or the difference between the smaller and the larger diameter of a piece, measured one inch or one foot apart, as the case may be. Suppose in Fig. 1 that the diameter at A is one inch, and the diameter at B, one and one-half inch, and that the distance or dimension between A and B is 12 inches or one foot. This piece, then, tapers one-half inch per foot, because the difference between the diameters at A and B is one-half inch. In Fig. 2, the diameter at C is 7/16 inch, and at D, 1/2 inch, and the distance between C and D is one inch. This piece, therefore, tapers 1/16 inch per inch. Tapers may also be expressed for other lengths than one inch and one foot. In Fig. 3, the diameter at E is 1 1/8 inch, and at F, 1 9/32 inch, and the dimension from E to F is 5 inches. This piece of work, therefore, tapers 5/32 inch in 5 inches, the difference between 1 9/32 and 1 1/8 being 5/32.

If the taper in a certain number of inches is known, the taper in 1 inch can easily be found. If the taper in 5 inches is 5/32 inch, the taper in 1 inch equals the taper in 5 inches divided by 5, or, in this case, $5/32 \div 5 = 1/32$, which is the taper per inch. The taper per foot is found by multiplying the taper per inch by 12. In this case, the taper per foot equals $12 \times 1/32 = 3/8$ inch. The length of the work is always measured parallel to the center line (axis) of the work, and never along the tapered surface.

The problems met with in regard to figuring tapers may be of three classes. In the first place we may have given us the figures for the large and small ends of a piece of work, and the length of the work, as in Fig. 4, and we want to find the taper per foot. In the second place we may know the diameter at one end, the length of the work, and the taper per foot, as in Fig. 5, and we want to find the diameter at the other end of the work. In the third place we may know the required diameters at both ends of the work, and the taper per foot, as in Fig. 6, and we want to find the dimension between the given diameters, or the length of the piece. We will now treat each of these problems in detail.

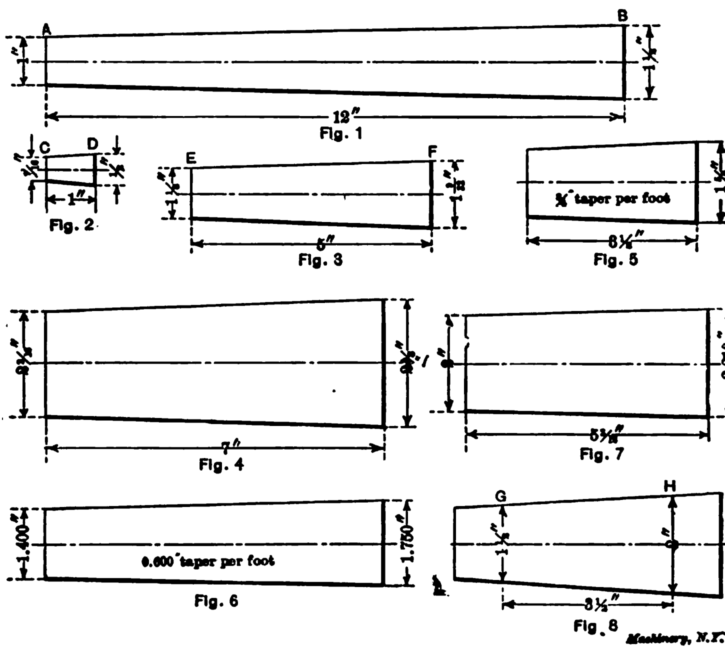
1. *To find the taper per foot when the diameters at the large and small ends of the work, and the length, are given.*

Referring to Fig. 4, the diameter at the large end of the work is 2 5/8 inches, the diameter at the small end, 2 3/16 inches, and the length of the work 7 inches. The taper in 7 inches is then equal to the difference between 2 5/8 inches and 2 3/16 inches, or 7/16 inch. The taper in one inch equals 7/16 divided by 7, or 1/16 inch; and the taper per foot is 12 times the taper per inch, or 12 times 1/16, which equals 3/4 inch. The taper per foot in Fig. 4, then, equals 3/4 inch.

If the dimension between the small and the large diameter is not expressed in even inches, but is 5 3/16 inches, for instance, as in Fig. 7, the procedure is exactly the same. Here the diameter at the large end is 2.216 inches and at the small end 2 inches. The taper in 5 3/16 inches is, therefore, 0.216 inch. This is divided by 5 3/16 to find the taper per inch.

$$0.216 \div 5\frac{3}{16} = 0.216 \div \frac{83}{16} = 0.216 \times \frac{16}{83} = 0.0416.$$

The taper per inch, consequently equals 0.0416 inch, and the taper per foot is 12 times this amount, or almost exactly 1/2 inch.



Figs. 1 to 8

Expressed as a formula, if all dimensions given are in *inches*, the previous calculation would take this form:

$$\text{taper per foot} = \frac{\text{large dia.—small dia.}}{\text{length of work}} \times 12.$$

It makes, of course, no difference if the large and small diameters are measured at the extreme ends of the work or at some other place on the work, provided the length or distance between the points where the diameters are given, is stated. In Fig. 8, the smaller and larger diameters are given at certain distances from the ends of the work, but the dimension from *G* to *H* is given, and the calculation is exactly the same as if the work were no longer than between *G* and *H*. The

following examples will tend to show how the figuring of the taper per foot enters in actual shop work.

Example 1.—Fig. 9 shows the blank for a taper reamer. The diameters at the large and small ends of the flutes, and the length of the fluted part, are stated on the drawing. It is required to find the taper per foot in order to be able to set the taper turning attachment of the lathe.

Referring to the figures given in Fig. 9, the difference in diameters at the large and small ends of the taper is 15/64 inch. This divided by the length of the flute, 7½ inches, gives us the taper per inch. This we find to be 1/32. The taper per foot is 12 times the taper per inch, or, in this case, then, ⅜ inch. The taper attachment of the lathe is, therefore, set to the ⅜-inch graduation, and the taper turned will be according to the diameters given on the drawing.

Example 2.—Fig. 10 shows a taper clamping bolt, entering into the design of a special machine tool. As seen from the cut, the drawing

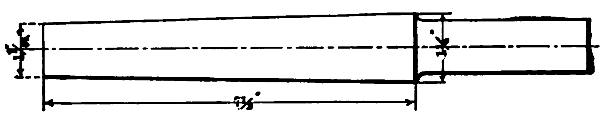


FIG. 9

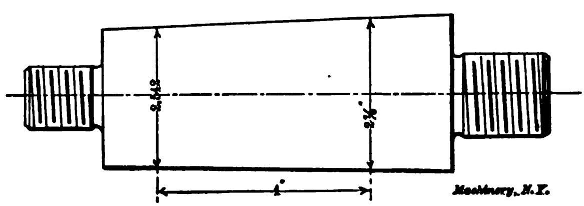


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FIG. 10

Figs. 9 and 10

calls for a diameter of 2⅞ inches a certain distance from the large end of the taper, and for a diameter of 2.542 inches a distance 4 inches further down on the taper. The taper in 4 inches is then 2⅞ inches minus 2.542 inches, or 0.333 inch. The taper in one inch equals this divided by 4, or 0.0833. The taper per foot is 12 times the taper per inch, or 12 times 0.0833, which equals one inch, almost exactly. The taper to which to turn the bolt in Fig. 10 is thus one inch per foot.

2. *If the diameter at one end of the taper is given, and also the length of the work and the taper per foot, to find the diameter at the other end of the work.*

Referring to Fig. 5, the diameter at the large end of the work is 1⅝ inch, the length of the work is 3½ inches, and the taper per foot is ¾ inch. We now want to find the diameter at the small end. In this case we simply reverse the method employed in our previous problems, where we wanted to find the taper per foot. In this case we know that the taper per foot is equal to ¾ inch. The taper in one

inch must be one-twelfth of this, or ¾ inch divided by 12, which equals 1/16 inch. Now, the taper in 3½ inches, which we want to find in order to know what the diameter is at the small end of the work, must be 3½ times the taper in *one* inch, or 3½ times 1/16, which equals 7/32. The taper in 3½ inches, then, is 7/32 inch, which means that the diameter at the small end of a piece of work, 3½ inches long, is 7/32 inch smaller than the diameter at the large end. The diameter at the large end, according to our drawing, is 1⅝ inch. The diameter at the small end, being 7/32 inch smaller, is therefore 1 13/32 inch.

Expressed as a formula, the previous calculation would take this form:

$$\text{dia. at small end} = \text{dia. at large end} - \left(\frac{\text{taper per foot}}{12} \times \text{length of work}\right)$$

If we now take a case where the diameter at the small end is given, as in Fig. 11, and the diameter at the large end is wanted, the figuring is exactly the same, except of course, we *add* the amount of taper in the length of the work to the small diameter to find the large diameter. When the large diameter is given, we *subtract* the amount of

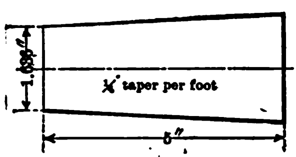


Fig. 11

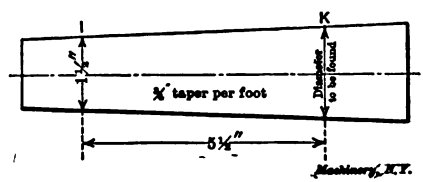


Fig. 12

taper in the length of the work to find the small diameter. This is so self-evident that no difficulties ought to be experienced on this account.

Referring again to Fig. 11, where the small diameter is given as 1.636 inch, the length of the work as 5 inches, and the taper per foot as ¼ inch, how large is the large diameter of the work? If the taper per foot is ¼ inch, the taper per inch is ¼ divided by 12 which equals 0.0208, and the taper in 5 inches consequently 5 times 0.0208, or 0.104 inch. The diameter at the large end of the work, which we are figuring, is, then, 0.104 inch larger than the diameter at the small end. The diameter at the small end is given on the drawing as 1.636 inch; adding 0.104 inch to this, we get 1.740 inch as the diameter at the large end.

Expressed as a formula, the previous calculation would take this form:

$$\text{dia. at large end} = \text{dia. at small end} + \left(\frac{\text{taper per foot}}{12} \times \text{length of work}\right)$$

It may again be well to call attention to the fact that it makes no difference whether the large and small diameters are figured at the extreme ends of the work or at some other points, as long as the

diameter to be found is located at one end of the length dimension, and the diameter stated on the drawing at the other. Thus, in Fig. 12 the diameter stated at *I* is given a certain distance up on the taper, and the diameter at *K*, which is wanted, is not at the end of the taper. But the dimension 5½ is given between the points *I* and *K* where these diameters are to be measured, and in figuring, one may reason as if the work ended at *I* and *K*, the diameter at *I* being the small diameter, the diameter at *K*, the large diameter, and 5½ inches the total length of the work. The following examples of direct practical application to shop work will prove helpful in remembering the principles outlined.

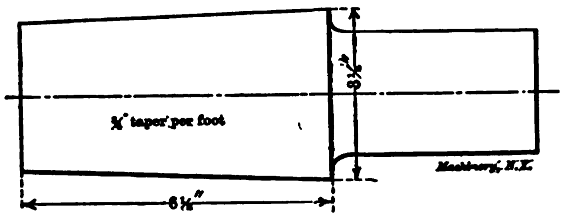


Fig. 13

Example 1.—Fig. 13 shows a taper tap, the blank for which is to be turned. The diameter at the large end of the threaded part is 3½ inches, as given on the drawing, the length of the thread is 6½ inches, and the taper per foot is ¾ inch. We want to find the diameter at the small end, in order to measure this end and ascertain that the tap blank has been correctly turned.

The taper per foot being 3/4 inch, the taper per inch is 3/4 divided by 12, or 1/16 inch. The taper in 6 1/2 inches is 6 1/2 times the taper in one inch, or 6 1/2 times 1/16 inch, which equals 13/32 inch. The taper in 6 1/2 inches being 13/32 inch means that the diameter at the small end of the tap blank is 13/32 inch smaller than the diameter at the

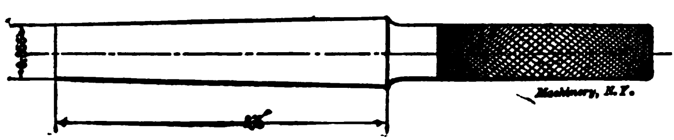


Fig. 14

large end. The diameter at the small end is, therefore, 3 3/32 inches.

Example 2.—Fig. 14 shows a taper gage for a standard Morse taper No. 1. The diameter at the small end is 0.356 inch, the length of the gage part is 2⅜ inches, and the taper per foot 0.600 inch. We want the diameter at the large end, in the first place in order to know what size stock to use for the gage, and later for measuring this diameter, when turned, to see that the taper turned is correct.

A taper of 0.600 per foot gives us a taper of 0.050 per inch. In 2⅜ inches the taper equals 2⅜ times 0.050, or 0.119 inch. This added to the diameter at the small end gives us the diameter at the large end: $0.356 + 0.119 = 0.475$ inch.

Example 3.—Fig. 15 shows a taper bolt used as a clamp bolt. The diameter 3¼ inches is given 3 inches from the large end of the taper. The total length of the taper is 10 inches. The taper is ⅜ inch per foot. We want to find the diameters at the extreme large and small ends of this piece.

We will first find the diameter at the large end. The taper per foot being ⅜ inch, the taper per inch equals 1/32 inch. The taper in 3 inches is consequently 3/32. This added to 3¼ inches will give us the diameter at the large end, which is 3 11/32 inches.

To find the diameter at the small end, subtract the taper in 10 inches, which is 10 times the taper in one inch, or 10 times 1/32, which equals 5/16, from the diameter 3 11/32 inches at the large end. This gives us the diameter at the small end 3 1/32 inches.

We can also find the diameter at the small end without previously finding the diameter at the extreme large end. The total length of the taper is 10 inches, and the dimension from where the diameter 3¼ inches is given to the large end is 3 inches. Consequently, the dimension from where the diameter 3¼ inches is given to the small end is 7 inches. The taper in one inch was 1/32 inch; in 7 inches, therefore,

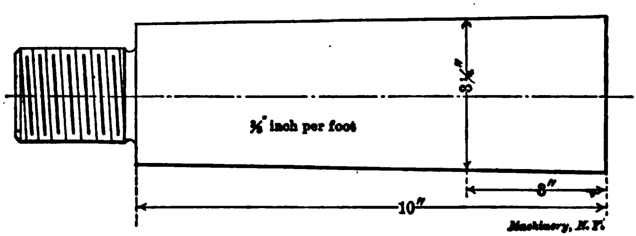


Fig. 15

7/32 inch. The diameter at the small end of the work is 7/32 inch smaller than 3¼ inches, or 3 1/32 inches, the same as found previously when we figured from the extreme large diameter of the taper.

3. *To find the distance between two given diameters on a tapered piece of work, if the taper per foot is known.*

Referring to Fig. 6, if the diameters at both ends of a tapered piece are known, together with the taper per foot, it is required to find the length of the work. Assume that the diameter at the large end of the piece is 1.750 inch, and at the small end, 1.400 inch. The taper per foot is 0.600 inch. How long is this piece of work required to be, in order to have the given diameters at the ends, with the taper stated? We know that the taper per foot is 0.600 inch. The taper per inch is then 0.600 divided by 12, or 0.050 inch. The difference in diameters between the large and the small ends of the work is 1.750 — 1.400, or 0.350 inch, which represents the taper in the length of the work. Now, we know that the taper is 0.050 inch in *one* inch. How many inches does it then require to get a taper of 0.350 inch? This we find by seeing how many times 0.050 is contained in 0.350, or, in other words, by dividing 0.350 by 0.050, which gives us 7 as answer. This means that it takes 7 inches for a piece of work to taper 0.350 inch,

if the taper is 0.600 per foot. The length of the work consequently is 7 inches in the case referred to.

Expressed as a formula the previous calculation would take the form:

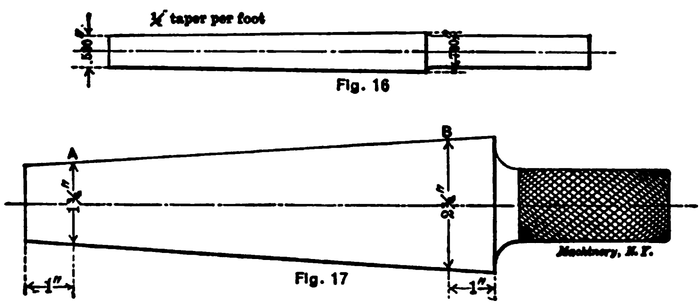
$$\text{length of work} = \frac{\text{dia. at large end} - \text{dia. at small end}}{\text{taper per foot} \div 12}$$

The taper per foot divided by 12, as given in the formula above, of course simply represents the taper per inch. The formula may therefore be written:

$$\text{length of work} = \frac{\text{dia. at large end} - \text{dia. at small end}}{\text{taper per inch}}$$

A few examples of the application of these rules will make their use in actual shop work clearer.

Example 1.—A taper reamer, Fig. 16, for standard taper pins, having ¼ inch taper per foot, is to be made. The diameter at the large end of the flutes is wanted to be 0.720 inch. The diameter at the point of the reamer must be 0.580 inch, in order to accommodate the longest



Figs. 16 and 17

taper pins of this size made. How long should the fluted part of the reamer be made?

The taper per foot is ¼ or 0.250 inch, and the taper per inch, consequently, 0.250 divided by 12, or 0.0208 inch. The taper in the length of reamer required is equal to the difference between the large and the small diameter, or 0.720 — 0.580 equals 0.140 inch. This amount of taper divided by the taper in one inch gives the required length of the flutes. Thus, 0.140, divided by 0.0208 equals 6.731, which represents the length of flutes required. This dimension is nearly 6¾ inches, and, being a length dimension of no particular importance, it would be made to an even fractional part of an inch.

Example 2.—In Fig. 17 is shown a taper master gage intended for inspecting taper ring gages of various dimensions. The smallest diameter of the smallest ring gage is 1¾ inch, and the largest diameter of the largest ring gage is 2¾ inches. The taper per foot is 1½ inch. It is required that the master gage extends one inch outside of the gages at both the small and the large ends, when these are tested. How long should the gage portion of this piece of work be?

The taper per foot is 1½ inch, which is equivalent to ⅛ inch taper per inch. The total taper from *A* to *B* in Fig. 17 is 2¾ minus 1¾, or one inch. Therefore, as the taper per inch, ⅛, is contained in the taper of one inch in the distance from *A* to *B* exactly 8 times, the dimension from *A* to *B* is 8 inches. The gage extends one inch beyond *A* and *B*, respectively, at either end, and the total length of the gage is, therefore, 10 inches.

Rules for Figuring Tapers

1. If the taper per foot is known, the taper per inch is found by *dividing the taper per foot by* 12.

2. If the taper per inch is known, the taper per foot is found by *multiplying the taper per inch by* 12.

3. To find the taper per foot, when the diameters at the large and small ends and the length of the taper are given, *substract the small diameter from the large, divide the remainder by the length of the taper, and multiply the quotient by* 12.

4. To find the diameter at the small end when the diameter at the large end, the length of the taper, and the taper per foot are given, *divide the taper per foot by* 12, *multiply the quotient by the length of the taper, and subtract the resulting dimension from the diameter at the large end.*

5. To find the diameter at the large end when the diameter at the small end, the length of the taper, and the taper per foot are given, *divide the taper per foot by* 12, *multiply the quotient by the length of the taper, and add the resulting dimension to the diameter at the small end.*

6. To find the dimension between two given diameters of a piece of work, when the taper per foot is given, *subtract the diameter at the small end from the diameter at the large end, and divide the remainder by the taper per foot divided by* 12.

7. To find how much a piece of work tapers in a certain length, when the taper per foot is given, *divide the taper per foot by* 12, *and multiply the quotient by the dimension of the certain length in which the taper is required.*

CHAPTER II

SETTING OVER TAIL-STOCK FOR TAPER TURNING

The live center and the tail center in a lathe are in alignment when a cutting tool, held in the tool-post of the lathe carriage, traverses in a direction parallel to a line connecting the points of the two centers; if a piece of work is then placed between the centers and revolved, and a cut taken over it, a cylindrical ("straight") piece will be turned. If the tail center is moved out of alignment with the live center an amount *A*, as shown in Fig. 18, then the center of the work at the tail center end will come nearer to the line of traverse *BC* of the tool than the center of the work at the live center end, and the diameter of the piece, when turned, will be smaller at the tail center than at the live center. It is therefore a common method to set over the tail-stock when a tapered piece is to be turned. The amount of the taper depends on the length of the work and the "set-over" of the tail center in each case.

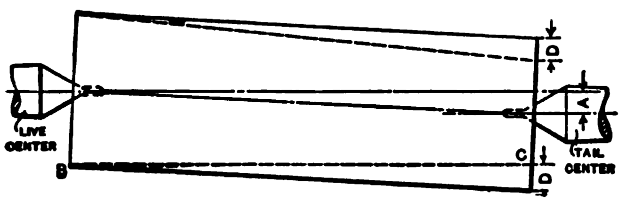


Fig. 18

When the tail center is set over an amount *A*, as shown in Fig. 18, the radius (one-half of the diameter) at the small end will be a dimension *D* smaller than the radius at the large end. This dimension *D* is also equal to the amount *A* which the tail center has been set over, and the taper of the work in the length between the centers, therefore, is two times the amount the tail-stock is set over; or, in other words, the tail-stock is set over one-half of the taper in the length of the work.

Taper per Foot and the Length are Known

The amount which the tail-stock must be set over can be determined if the taper per foot of the work and the length are known.

Assume that a piece of work, 7½ inches long, is required to be turned with a taper per foot equal to ¾ inch. We must first know how much the work tapers in 7½ inches. This we find by dividing ¾ by 12, and multiplying the quotient by 7½ (see page 12, Rule 7).

$$(\tfrac{3}{4} \div 12) \times 7\tfrac{1}{2} = 15/32.$$

The taper in 7½ inches thus is 15/32 inch, and as the tail-stock is moved one-half of this, it is set over 15/64 inch.

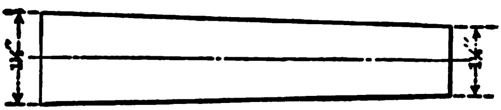
When the taper per foot and the length of the work are given, we can calculate the amount to set over the tail-stock from the following formula:

$$\text{amount to set over tail-stock} = \frac{1}{2} \times \left(\frac{\text{taper per foot}}{12} \times \text{length of work}\right)$$

Expressed in words, this formula reads:

To find the amount to set over the tail-stock when the taper per foot and the length of the work are known, *divide the taper per foot by* 12, *multiply the quotient by the length of the work, and divide the result by* 2. (To divide by 2 is the same as to multiply by ½.)

Owing to the fact that the work is not supported by the lathe centers at its extreme ends, but that the lathe centers enter into the work and support it at points a short distance from the ends, it is not practicable to calculate the amount to set over the tail-stock so definitely that the taper can be turned to exact dimensions without a trial cut; but the calculation for setting over the tail-stock gives a close approximation,



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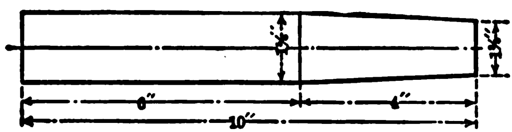


Machinery,N.Y.

Fig. 20

and when a trial cut on the work has been taken, the final adjustment of the tail-stock to obtain the correct taper can be easily made.

Method Used when the Diameters at Both Ends of a Tapered Piece are Known

If the diameters at both the large and small ends of work tapering for its full length, are given, the amount to set over the tail-stock can be determined without knowing the taper per foot, because all that is necessary to know is the taper in the length between the centers of the lathe. If, for instance, the diameter at the large end of the work is 1½ inch and the diameter at the small end 1¼ inch, as shown in Fig. 19, the amount to set over the tail-stock will be one-half of the difference between the large and small diameters, or ⅛ inch. When the diameters at the large and small ends are known, the following formula is therefore used:

$$\text{amount to set over tail-stock} = \frac{1}{2} \times (\text{large diameter} - \text{small diameter}).$$

Expressed in words, this formula reads:

To find the amount to set over the tail-stock for work tapering for its full length, when the diameters at the large and small ends are known, *subtract the small diameter from the large, and divide the remainder by* 2.

Method Used when Part of the Work is Turned Straight and Part Tapered

If part of the work is turned straight and part of it turned tapered, as shown in Fig. 20, the taper in the whole length of the work must be determined, and then the tail-stock set over one-half of this amount. In Fig. 20 the work shown is $1\frac{3}{8}$ inch at the small end of the taper. It is tapered for 4 inches, and the diameter at the large end of the taper is $1\frac{5}{8}$ inch. It is then turned straight for the remaining 6 inches, the total length being 10 inches. We must first find what the taper would be in 10 inches if the whole piece had been tapered with the same taper as now required for 4 inches. The taper in 4 inches is $1\frac{5}{8} - 1\frac{3}{8} = \frac{1}{4}$ inch. The taper in 1 inch, consequently, is 1/16 inch, and in 10 inches, $10 \times 1/16 = \frac{5}{8}$ inch. The amount to set over the tail-stock is one-half of this, or 5/16 inch.

If, in a case as shown in Fig. 20, the diameter at the small end is not given, but the taper per foot of the tapered part stated instead,

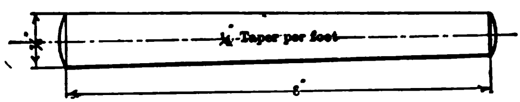


Fig. 21

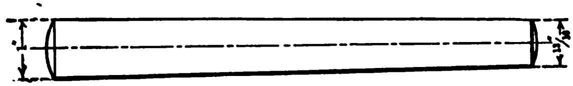


Fig. 22

the taper in the total length of the work can be found directly; if the taper per foot be $\frac{3}{4}$ inch, the taper in 10 inches is $(\frac{3}{4} \div 12) \times 10 = \frac{5}{8}$ inch. (See page 12, Rule 7.) The amount to set over the tail-stock, consequently, is 5/16 inch. The following formula is used when part of the work is turned straight and part tapered:

$$\text{amount to set over tail-stock} = \frac{1}{2} \times \left(\frac{\text{taper per foot}}{12} \times \text{total length of work}\right)$$

Expressed as a rule, this formula would read:

To find the amount to set over the tail-stock for work partly tapered and partly straight, when the taper per foot and the total length of the work are known *divide the taper per foot by* 12, *multiply the quotient thus obtained by the total length of the work, and divide by* 2.

If the taper per foot is not given, it must be found before using this formula and rule. (See page 12, Rule 3.)

The following examples will help to give a clear idea of the application of these rules.

Example 1.—The taper pin shown in Fig. 21 is 8 inches long, and tapers $\frac{1}{4}$ inch per foot. How much should the tail-stock be set over when turning this pin?

Dividing the taper per foot by 12 gives us 0.0208. Multiplying this figure (which represents the taper per inch) by 8 gives us 0.166 as

the taper in 8 inches. Dividing this by 2 gives us the amount required to set over the tail-stock. This amount then is 0.083 inch.

Example 2.—Another taper pin, Fig. 22, is 1 inch in diameter at the large end, and 13/16 inch at the small end. How much should the tail-stock be set over for turning this pin?

The total taper of this pin is found by subtracting the diameter at the small end, 13/16 inch, from the diameter at the large end, 1 inch. This gives us a remainder of 3/16. One-half of this amount, or 3/32 inch, represents the amount which the tail-stock should be set over.

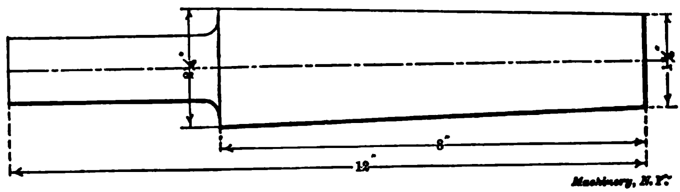


Fig. 23

Example 3.—A taper gage, as shown in Fig. 23, is to be turned by setting over the tail-stock. The diameter at the large end of the taper is 2¼ inches, the diameter at the small end is 1¾ inch, the length of the taper, 8 inches, and the total length, 12 inches. How much should the tail-stock be set over?

Subtracting the diameter at the small end, 1¾ inch, from the diameter at the large end, 2¼ inches, gives us a taper of ½ inch in 8 inches. Dividing ½ by 8, gives us the taper in one inch, which is 1/16 inch. Multiplying this by the total length of the work, 12 inches, gives us ¾ inch, which, divided by 2, gives us, finally, the required amount which the tail-stock is to be set over. This latter is, therefore, set over ⅜ inch.

CHAPTER III

CUTTING SPEEDS AND FEEDS

The cutting speed of a tool is the distance in feet which the tool point cuts in one minute; thus, if the point of a lathe tool cuts 40 feet, measured around the work, on the surface of a casting turned in the lathe, in one minute, we say that the cutting speed is 40 feet per minute.

On the planer, the cutting speed is equal to the length of cut that would be taken in one minute. If a cut 12 feet long is taken in 20 seconds, then, as 20 seconds is one-third of a minute, a cut 36 feet long could be made with the same speed in one minute, and the cutting speed is 36 feet per minute.

When drilling a hole in the drill press, the cutting speed is the number of feet that the outer corners of the cutting edges travel in one minute.

Cutting Speeds in the Lathe, Boring and Turning Mill and Drill Press

The problems in regard to cutting speeds in the lathe or turning and boring mill may be divided in two groups.

1. *The diameter of the work turned in a lathe or boring mill and the required cutting speed are known. How many revolutions per minute should the work make?*

Assume that the diameter of the work is 5 inches, and the required cutting speed 40 feet per minute. When the diameter of the work is known, its circumference equals the diameter times 3.1416. Therefore the circumference of the work in this case is $5 \times 3.1416 = 15.708$ inches. For calculations of this character it will be near enough to say that the circumference is 15.7 inches. For each revolution of the work, the length of its circumference passes the tool point once; thus for each revolution, a length of 15.7 inches passes the tool. As the cutting speed is expressed in feet, the length 15.7 inches should also be expressed in feet, which is done by dividing by 12, thus obtaining $15.7 \div 12 = 1.308$ foot, as the circumference of the work. The next question is, how many revolutions, each equivalent to 1.308 foot, does it require to get a cutting speed of 40 feet. This we get by finding how many times 1.308 is contained in 40, or, in other words, by dividing 40 by 1.308. The quotient of this division is 30.6. Therefore, 30.6 revolutions per minute are required to obtain a cutting speed of 40 feet per minute in this case.

This calculation is expressed by the formula:

$$\text{revolutions per minute} = \frac{\text{cutting speed in feet per minute}}{\left(\begin{array}{c}\text{diameter of}\\ \text{work in inches}\end{array} \times 3.1416\right) \div 12}$$

If instead of turning work 5 inches in diameter, a hole 5 inches in diameter is to be bored by an ordinary forged boring tool or a tool inserted into a boring bar, and the cutting speed is required to be 40 feet per minute, the calculation for the revolutions per minute is carried out in the same manner as mentioned before, and the same formula is used, except that in the formula we write "diameter of hole to be bored in inches" instead of "diameter of work in inches."

For work done in the drill press, the formula can also be used by substituting "diameter of hole to be drilled in inches" for "diameter of work in inches."

2. *The number of revolutions which the work makes in a lathe or boring mill, and the diameter are known. What is the cutting speed?*

A brass rod one inch in diameter is being turned. By counting the number of revolutions of the spindle of the lathe by means of a speed indicator (instrument for counting the number of revolutions of revolving shafts or spindles) it is found that the work revolves 382 revolutions per minute. To find the cutting speed, the circumference

of the work is first figured and changed into feet. The circumference in inches is $1 \times 3.1416 = 3.1416$, and $3.1416 \div 12 = 0.262$, the circumference in feet, or the distance passed over by the tool point for each revolution. During 382 revolutions, the distance passed over is $382 \times 0.262 = 100$ feet, which thus is the cutting speed per minute.

This calculation is expressed by the formula:

$$\text{cutting speed in feet per minute} = \frac{\text{dia. of work in inches} \times 3.1416}{12} \times \text{revolutions per minute}$$

If in this formula "diam. of work in inches" is substituted by "diameter of bored or drilled hole in inches," the formula can be used for cutting speeds of drills and boring tools also.

(If the cut taken on a piece being turned is deep in proportion to the diameter of the work, it is preferable in calculations for the cutting speed and revolutions per minute to consider the *mean* diameter of the cut instead of the outside diameter of the work, and use the value for the mean diameter in the formulas given. When the outside diameter and the depth of the cut are known, the mean diameter equals the outside diameter minus the depth of cut.)

Cutting Speeds of Milling Cutters

The cutting speeds of milling cutters can be calculated when the diameter of the cutter and the revolutions per minute are given. For instance, the diameter of a cutter is 6 inches and it makes 40 revolutions per minute. To find the cutting speed in feet per minute, first find the circumference of the cutter; thus, $6 \times 3.1416 = 18.8496$, or about 18.8 inches; change this to feet; thus, $18.8 \div 12 = 1.566$ feet. As the cutter makes 40 revolutions per minute, the cutting speed is $40 \times$ the circumference, or $40 \times 1.566 = 62.64$ feet per minute.

If, in the formula just given above, "diam. of work in inches" is substituted by "diameter of cutter," this formula can be used for finding the cutting speed of milling cutters.

If the required cutting speed of a cutter is given and its diameter known, and the number of revolutions at which it should be run is to be found, the formula on page 17 can be used, in this case, also, of course, "diameter of work in inches" being substituted by "diameter of cutter."

Rules for Calculating Cutting Speeds

1. *To find the number of revolutions per minute when the diameter of the work turned, the hole drilled or bored, or the milling cutter used, in inches, and the cutting speed in feet per minute are given, multiply the diameter by* 3.1416 *and divide the result by* 12. *Then divide the given cutting speed by the quotient thus obtained.*

2. *To find the cutting speed in feet per minute when the diameter of the work to be turned, the hole drilled or bored, or the milling cutter used is given in inches, and the number of revolutions per minute are known, multiply the diameter by* 3.1416 *and divide the result by* 12. *Then multiply the quotient thus obtained by the number of revolutions per minute.*

Average Cutting Speeds

The cutting speed to use when cutting metals depends primarily upon the kind of tool used and the metal being cut. It is not possible to state exactly what the correct speed would be for all different cases, but the speeds in the table below are given as embodying good average practice when ordinary carbon steel tools are used.

For high-speed steel tools, these speeds may be doubled. In starting high-speed steel drills, a cutting speed of about 50 to 70 feet per minute, for machine steel, 60 to 80 feet per minute for cast iron, and 100 to 140 feet for brass, is recommended. When a few holes have been drilled at these speeds, still higher speeds may be employed.

Feed of Cutting Tools

The feed of a lathe tool is its sidewise motion (traverse) for each revolution of the work; thus, if the feed is 1/32 inch it means that for each revolution of the work, the lathe carriage and tool move 1/32 inch along the lathe bed, thus cutting a chip 1/32 inch wide.

The feed of a drill in the drill press is the downward motion of the drill per revolution. The feed of a milling cutter is the forward movement of the milling machine table for each revolution of the cutter.

TABLE OF CUTTING SPEEDS IN FEET PER MINUTE

Machine	Material			
	Tool Steel Annealed	Wrought Iron and Machine Steel	Cast Iron	Brass
Lathe, Planer and Shaper	18 to 25	30 to 40	40 to 50	80 to 125
Milling Machine..........	25 to 35	35 to 45	40 to 60	80 to 120
Drill Press................	20	30	35	60

Sometimes the feed is expressed as the distance which the drill or the milling machine table moves forward in one minute. In order to avoid confusion, it is, therefore, always best to state plainly in each case whether feed per revolution or feed per minute is meant.

Time Required for Turning Work in the Lathe

The most common calculations in which the feed of a lathe tool enters is the time required for turning or boring a given piece of work, when the feed, cutting speed and diameter of work (or the number of revolutions per minute) are known.

Assume that a tool steel arbor, 2 inches in diameter, is to be turned. The length to be turned on the arbor (the length of cut) is 10 inches. The cutting speed is 25 feet per minute and the feed or traverse of the cutting tool is 1/32 inch per revolution. How long a time would it require to take one cut over the surface of the work? We first find the number of revolutions per minute of the work, which equals

$$\frac{25}{(2 \times 3.1416) \div 12} = \frac{25}{0.524} = 47.7.$$

As the tool feeds forward 1/32 inch for each revolution of the work, it is fed forward 47.7 × 1/32 or 1.49 inch in one minute. The time required to traverse the whole length of the work, 10 inches, is obtained by finding how many times 1.49 is contained in 10, or, in other words, by dividing 10 by 1.49. The quotient of this division is 6.71 minutes. It would thus take 7 minutes, approximately, to traverse the work once with the cutting speed and feed given.

Expressed as a formula the calculation takes this form:

$$\text{time required for one cut over the work} = \frac{\text{length of cut}}{\text{rev. per min.} \times \text{feed per revolution}}$$

Expressed as a rule the formula would be:

To find the time required to take one complete cut over a piece of work in the lathe when the feed per revolution, the total length of cut, and the number of revolutions per minute are given, divide the total length of the cut by the number of revolutions per minute multiplied by the feed per revolution.

If the cutting speed and diameter of work are given instead of the number of revolutions, first find the revolutions per minute before applying the formula or rule above. (The number of revolutions per minute is found by Rule 1 on page 18.)

When the feed per revolution is known, the feed per minute equals the revolution per minute times the feed per revolution.

CHAPTER IV

SCREW THREADS AND TAP DRILLS

The terms pitch and lead of screw threads are often confused. The pitch of a screw thread is the distance from the top of one thread to the top of the next thread, as shown in Fig. 24. No matter whether the screw has a single, double, triple or quadruple thread, the pitch is always the distance from the top of one thread to the top of the next thread. The lead of a screw thread is the distance the nut will move forward on the screw, if it is turned around one full revolution. In the *single-threaded screw, the pitch and lead are equal*, because the nut would move forward the distance from one thread to the next, if turned around once. In a double-threaded screw, however, the nut will move forward two threads, or twice the pitch, so that in a double-threaded screw, the lead equals twice the pitch. In a triple-threaded screw, the lead equals three times the pitch, and so forth. The lead may also be expressed as being the distance from center to center of the *same* thread, after one turn, as indicated in Fig. 25, which shows the pitch and the lead for three screws with Acme threads, the first single-threaded, the second double-threaded, and the last, triple-threaded. In a single-threaded screw, the lead is the distance to the

next thread from the one first considered. In a double-threaded screw there are two threads running side by side around the screw, so that the lead is here the distance to the second thread from the one first considered. In a triple-threaded screw, it is the distance to the third thread, and so forth.

The word pitch is often, though improperly, used in the shop to denote *number of threads per inch.* We hear of screws having 12

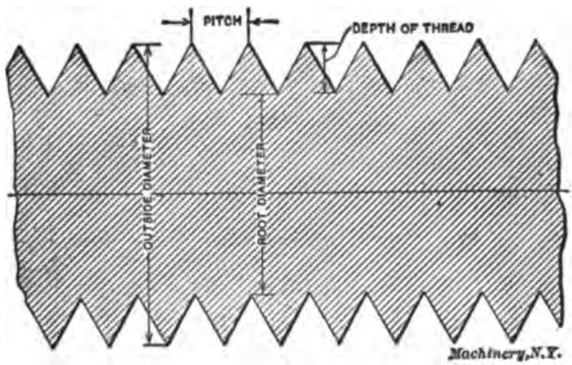


Fig. 24

pitch thread, 16 pitch thread, when 12 threads per inch and 16 threads per inch is what is really meant.

The number of threads per inch is the number of threads counted in the length of one inch, if a scale is held against the side of the screw, and the threads counted as shown in Fig. 26. If there is not a whole number of threads in one inch, count the threads in two or more inches, until the top of one thread comes opposite an inch-mark, and

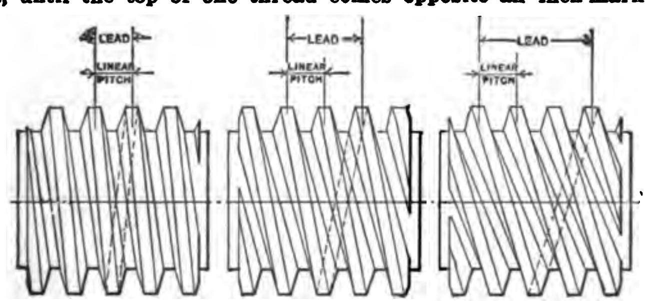


Fig. 25

then divide by the number of inches to find the number of threads in one inch, as shown in Fig. 27.

The number of threads per inch equals 1 divided by the pitch, or expressed as a formula:

$$\text{number of threads per inch} = \frac{1}{\text{pitch}}$$

The pitch of a screw equals 1 divided by the number of threads per inch, or

$$\text{pitch} = \frac{1}{\text{number of threads per inch}}$$

Thus, if the number of threads per inch equals 16, the pitch equals 1/16. If the pitch equals 0.05, the number of threads per inch equals $1 \div 0.05 = 20$. If the pitch equals 2/5 inch, the number of threads per inch equals $1 \div 2/5 = 2\frac{1}{2}$.

Confusion is often caused by indefinite designation of multiple-thread (double, triple, quadruple, etc.) screws. One way of expressing that a double-thread screw is required is to say, for instance: "3 threads per inch double," which means that the screw is cut with 3 *double* threads, or 6 threads per inch, counting the threads by a scale placed alongside of the screw, as shown in Fig. 26. The pitch of this screw

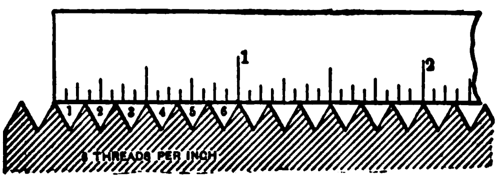


Fig. 26

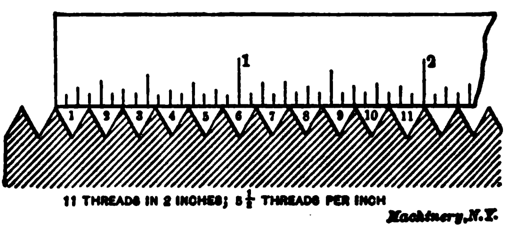


Fig. 27

thus is 1/6 inch, and the lead twice this, or 1/3 inch. To cut this screw, the lathe will be geared to cut 3 threads per inch, but the thread will be cut only to the depth required for 6 threads per inch. "Four threads per inch triple" means that there are 4 times 3, or 12 threads along one inch of the screw, when counted by a scale; the pitch of the screw is 1/12 inch, but being a triple screw, the lead of the thread is 3 times the pitch, or ¼ inch.

The best way of expressing that a multiple-thread screw is to be cut, when the lead and the pitch have been figured, is, for example: "1/4 inch lead, 1/12 inch pitch, triple thread." In the case of single-threaded screws, the number of threads per inch and the form of the thread only are given. The word "single" is not required.

There are three standard threads in common use in American shops. These are shown in Fig. 28, and are the United States standard thread, the sharp V-thread, and the Acme standard thread.

United States Standard Thread

This thread is the most commonly used thread form for all ordinary screws. The thread is provided with a small flat at the top and at the bottom of the thread, as shown in the illustration to the left in Fig. 28.

The depth of the United States (U.S.) standard thread equals 0.6495 × pitch. The width of the flat of the thread at the bottom and top equals ⅛ × pitch. The root diameter is found by subtracting two times the depth of the thread from the outside diameter of the screw.

[The root diameter of a screw thread is the diameter at the bottom of the thread, as shown in connection with the V-thread in Fig. 24.]

Standard Sharp V-Thread

This thread has no flat at the top or at the bottom, but the sides of the thread form a sharp point, as shown in the illustration, Fig. 28. The depth of the thread equals 0.866 × pitch. The root diameter is

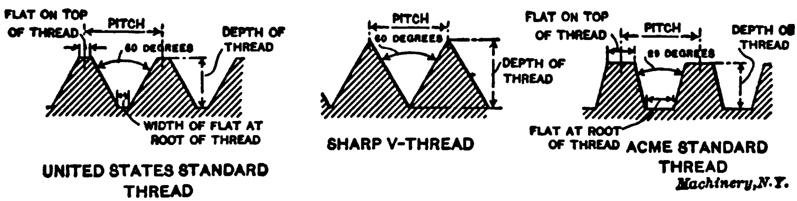


Fig. 28

found by subtracting two times the depth of the thread from the outside diameter of the screw.

Acme Standard Thread

The form of the Acme standard thread is shown to the right in Fig. 28. The depth of the thread equals ½ × pitch + 0.010 inch. The flat at the top of the thread equals 0.3707 × pitch. The width of the flat at the root of the thread equals 0.3707 × pitch — 0.0052 inch. The root diameter of the thread, of course, is found as before, by subtracting two times the depth of the thread from the outside diameter of the screw.

Tap Drill Sizes

The tap drills used for drilling holes previous to tapping are usually somewhat larger in diameter than the root diameter of the thread.

The tap drill diameter for ordinary work for the United States standard thread equals the root diameter + (⅛ × pitch).

The tap drill diameter for sharp V-thread equals the root diameter of the thread + (¼ × pitch).

The tap drill diameter for Acme standard thread equals the root diameter + 0.020 inch.

A table of double depths of threads for U. S. and sharp V-threads is given on the following page. The figures in this table opposite any given number of threads per inch are simply subtracted from the outside diameter of the screw, to obtain the root diameter.

Example: Find the root diameter of a 1 5/16 inch diameter screw having 8 U. S. threads per inch. The diameter of the screw in decimals is 1.3125. Subtract from this the double depth of the thread, or the so-called "constant," in the table, which is 0.1624, as given opposite 8 threads per inch. The remainder, 1.1501, is the root diameter of the thread.

TABLE OF DOUBLE DEPTH OF SCREW THREADS

Threads per Inch	U. S. Standard Thread	Standard V-Thread	Threads per Inch	U. S. Standard Thread	Standard V-Thread
2 1/4	0.5774	0.7698	18	0.0722	0.0962
2 3/8	0.5470	0.7293	20	0.0650	0.0866
2 1/2	0.5196	0.6928	22	0.0590	0.0787
2 5/8	0.4949	0.6598	24	0.0541	0.0722
2 3/4	0.4724	0.6298	26	0.0500	0.0666
2 7/8	0.4518	0.6025	28	0.0464	0.0619
3	0.4330	0.5774	30	0.0433	0.0577
3 1/4	0.3997	0.5329	32	0.0406	0.0541
3 1/2	0.3712	0.4949	34	0.0382	0.0509
4	0.3248	0.4330	36	0.0361	0.0481
4 1/2	0.2887	0.3849	38	0.0342	0.0456
5	0.2598	0.3464	40	0.0325	0.0433
5 1/2	0.2362	0.3149	42	0.0309	0.0412
6	0.2165	0.2887	44	0.0295	0.0394
7	0.1856	0.2474	46	0.0282	0.0377
8	0.1624	0.2165	48	0.0271	0.0361
9	0.1443	0.1925	50	0.0260	0.0346
10	0.1299	0.1732	52	0.0250	0.0333
11	0.1181	0.1575	56	0.0232	0.0309
12	0.1083	0.1443	60	0.0217	0.0289
13	0.0999	0.1332	64	0.0203	0.0271
14	0.0928	0.1237	68	0.0191	0.0255
15	0.0866	0.1155	72	0.0180	0.0241
16	0.0812	0.1083	80	0.0162	0.0217

These constants are subtracted from the outside diameter of the tap or screw; the result is the root diameter of the thread

When the table of double depth of threads is used for multiple-thread screws, it should be noted that the number of threads per inch as measured along the screw as shown in Fig. 26, should be taken. Thus, for instance, to find the root diameter of a screw having 6 threads per inch double, the figures opposite 12 threads (6 double) should be used.

CHAPTER V

TRAINS OF GEARS

Suppose that two shafts *A* and *B*, as shown in Fig. 29, are to be connected by gearing so that shaft *A* makes one revolution while shaft *B* makes three. To obtain this result, the gear on *A* must have three times as many teeth as the gear on *B*. Assume that we have 90 teeth in the gear on *A*; the gear on *B* must then have only 30 teeth. Each

time the small gear on *B* turns around one complete revolution, it engages 30 teeth in the gear on shaft *A*. It therefore must turn around three full revolutions in order to engage all the 90 teeth in the larger gear, or, in other words, the smaller gear revolves three times in order to revolve the gear on *A* once. The *ratio* of this gearing is 3 to 1. If one gear revolves three times while the mating gear revolves four times, the ratio of the gearing would be 3 to 4. The ratio of gearing expresses the relation between the number of times which one gear revolves and the number of times the mating gear revolves in the same time.

Effect of Idlers

Assume that the shafts *C* and *D* in Fig. 30 are required to run in a ratio of 5 to 1, that is, shaft *C* is to revolve five times while shaft *D*

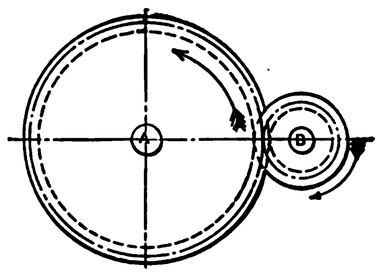


Fig. 29

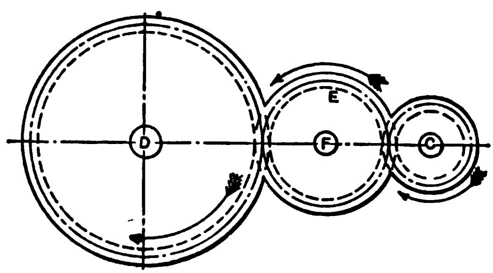


Fig. 30

revolves once. If a 20-tooth gear is placed on shaft *C*, the gear on shaft *D* must have five times as many teeth, or 100 teeth; then it will revolve but once when the gear on *C* revolves five times. Assume that we put gears with the number of teeth mentioned on the shafts. The distance between the shafts may now be such that the gears will not mesh or engage. An intermediate gear *E* mounted on a stud *F* may then be placed so that it meshes with both the gear on *C* and the gear on *D*. The intermediate gear simply transmits motion from the gear on *C* to the gear on *D*, but has no influence on the ratio of speed of the shafts *C* and *D*. When the intermediate gear is in place, the gear on *C* still revolves five times while the gear on *D* revolves once.

If we place a number of intermediate gears, *E*, *F*, and *G*, in the train, as in Fig. 32, the result would still be the same, the gear on *C*

would turn 5 times while the gear on *D* turned once, as long as the number of teeth in the gear on *D* is 5 times the number of teeth in the gear on *C*.

In order to prove this, let us assume that in Fig. 32, the gear on stud *C* has 20 teeth, and the gear on stud *D*, 100 teeth, so that consequently the stud *C* makes 5 revolutions, while stud *D* makes one.

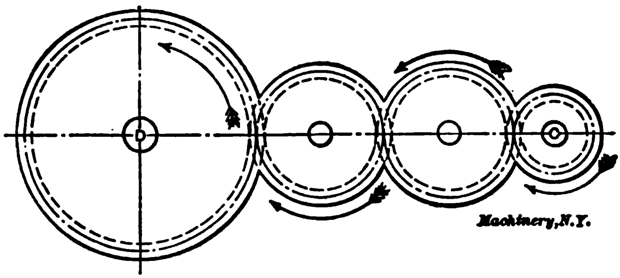


Fig. 31

The intermediate gears, *E*, *F*, and *G*, have 50, 40, and 40 teeth, respectively, as shown in the cut. Now, when the gear on *D* turns around once, the gear *G* must turn 2½ times (100/40 = 2½). The gear *F*, having the same number of teeth as gear *G*, makes one revolution while *G* makes one, and consequently also turns 2½ times while the gear on *D* turns once. The gear on *E*, having 50 teeth, turns 4/5 of a

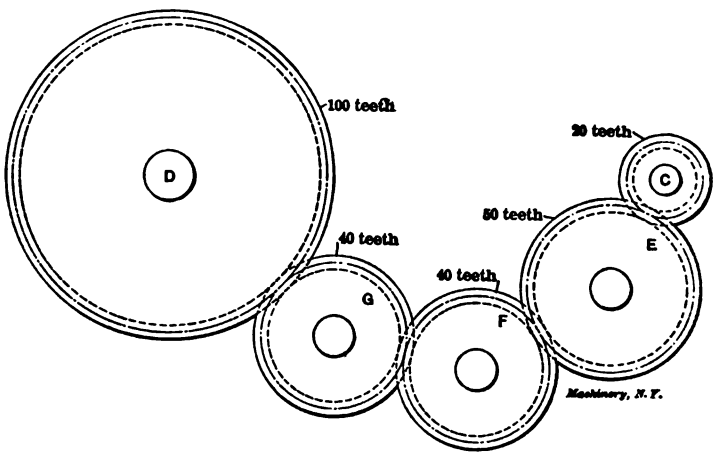


Fig. 32

revolution while gear *F* revolves once (40/50 = 4/5), and consequently, while *F* makes 2 1/2 revolutions, gear *E* makes 2 1/2 × 4/5 = 5/2 × 4/5 = 2 revolutions. Thus *E* turns twice while the gear on stud *D* revolves once. Finally, the gear on *C* turns 2 1/2 times to each revolution of gear *E* (50/20 = 2 1/2), or 5 times to 2 revolutions of *E*. But 2 revolutions of *E* correspond, as we have seen, to one revolution of

the gear on stud D; consequently, the gear on stud C makes 5 revolutions to one of the gear on stud D, which, as we previously said, is also the case if these two gears had been connected directly without any intermediate gearing.

The intermediate gears, however, affect the direction in which the gear on D revolves. In Fig. 29, when the gear on B revolves in a right-hand direction (in the same direction as the hands of a watch), the gear on A, which is driven by it, will move in a left-hand direction (in a direction opposite to that of the hands of a watch. Thus when there is no intermediate gear, the driving gear and the driven gear revolve in opposite directions. In Fig. 30, again, the gear on C moves in a right-hand direction, the gear on F in a left-hand direction, and the gear on D in a right-hand direction, so that in this case both the driver on C and the driven gear on D move in the same direction.

If there be two idlers, as shown in Fig. 31, the speed ratio between the shafts C and D still remains the same as before, but it will be seen from the arrows indicating the directions in which the different gears revolve, that in this case, the driver on C and the driven gear on D move in opposite directions. The use of two idlers, therefore, makes the driven gear run in the same direction as if there was no idler between the gears on C and D. If there were three idlers the gears on C and D would run in the same direction, and if there were four idlers, they would run in opposite directions, and so on.

Gears Required for a Given Speed Ratio

If we have two gears A and B, as shown in Fig. 33, and the ratio of the speed of gear A to the speed of gear B and the number of teeth in one of the gears are given, the number of teeth required in the other gear can be determined; and if the ratio only is given, the number of teeth in both the gears can be found.

Assume that the ratio of the speed of gear A to gear B is 1 to 4. This means that gear A revolves once while gear B revolves four times. If there be 80 teeth in gear A, gear B must have one-fourth of this number, or 20 teeth. Had the speed ratio been 2 to 5, then if gear A had 50 teeth, gear B would have $\frac{2}{5} \times 50 = 20$ teeth. From this we may formulate the following rule:

If the speed ratio of the driving gear to the driven gear and the number of teeth in the driving gear are given, the number of teeth in the driven gear may be found by multiplying the number of teeth in the driving gear by the speed ratio, written as a fraction.

Assume, for instance, that the speed ratio of gear A to gear B is 3 to 5, and that the number of teeth in gear A is 60. Writing the speed ratio as a fraction gives us 3/5, and this multiplied by 60 gives us 36, which is the number of teeth in the gear B.

Note that when the speed ratio of gear A to gear B is given, we multiply the number of teeth in gear A with the speed ratio written as a fraction, to get the number of teeth in gear B. But if the number of teeth in gear B is given, and the number of teeth in gear A is to

be found, the number of teeth in gear *B* should be multiplied by the *inverted* speed ratio. (To invert a fraction means to turn it upside down, so that the numerator becomes the denominator, and the denominator becomes the numerator.) If the speed ratio of gear *A* to gear *B* is 2 to 3, and the number of teeth in gear *B* is 34, then the number of teeth in gear *A* is found by multiplying 34 by the inverted ratio; thus:

$$34 \times \frac{3}{2} = \frac{102}{2} = 51.$$

It is helpful to remember when doing calculations of this kind that the larger gear, which, of course, is the gear with the greater number of teeth, is always the gear which will make the fewer number of revolutions. Keeping this in mind will prevent mistakes.

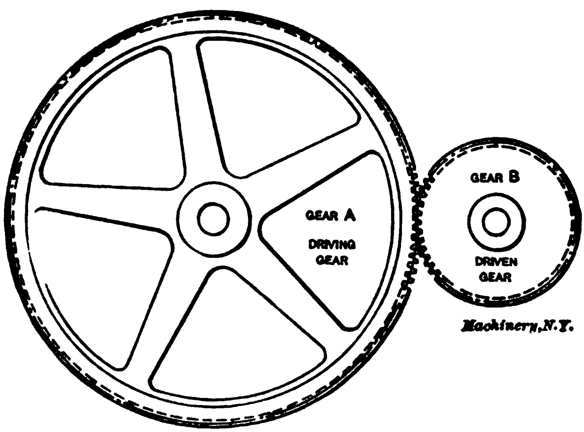


Fig. 33

If the speed ratio of two shafts is given, and the numbers of teeth for two gears which will transmit motion between the shafts at the given speed ratio is to be found, write the speed ratio in the form of a fraction and multiply the numerator and the denominator by the same number until a new fraction is obtained, having numerator and denominator expressing suitable numbers of teeth for the gears.

If the given speed ratio be 4 to 7, and it is required to find two gears which will transmit motion between the two shafts at this ratio, write the ratio as a fraction (4/7), and multiply numerator and denominator by the *same* number, thus:

$$\frac{4}{7} = \frac{4 \times 8}{7 \times 8} = \frac{32}{56}.$$

By multiplying 4 and 7 by 8 we obtain two gears with 32 and 56 teeth, which will transmit the motion from one shaft to another at the required ratio. The larger gear, with 56 teeth is placed on the shaft with the lowest number of revolutions.

Finding the Number of Revolutions per Minute

If the numbers of teeth in the two gears and the number of revolutions per minute of one shaft are known, and it is required to find the number of revolutions of the other shaft, the following rule is used:

The number of revolutions of one shaft multiplied by the number of teeth of the gear on the same shaft, divided by the number of teeth in the gear on the second shaft, gives the number of revolutions of the second shaft.

Written as a formula this rule would be:

$$\frac{\text{rev. of first shaft} \times \text{teeth in gear on first shaft}}{\text{teeth in gear on second shaft}} = \text{rev. of second shaft.}$$

If the speed ratio is given instead of the numbers of teeth, the numbers of teeth may first be found by the rules previously given.

Finding the Speed Ratio

The speed ratio can be found if either the numbers of teeth in both of the gears, or the numbers of revolutions per minute of both of the gears, are given.

If the numbers of revolutions of two gears A and B are given, the ratio of the speed of gear A to gear B is found by placing the number of revolutions of gear A as the numerator and the revolutions of gear B as the denominator of a fraction, and reducing this fraction to its lowest terms.* If gear A makes 34 revolutions, and gear B, 85 revolutions, the speed ratio of gear A to gear $B = \frac{34}{85} = \frac{2}{5}$ or, as it is commonly expressed, 2 to 5.

If the numbers of teeth in the gears A and B are given, the speed ratio of gear A to gear B is found by writing the number of teeth of gear B as the numerator, and of gear A as the denominator of a fraction, and reducing the fraction to its lowest terms. If the number of teeth in gear A is 60, and in gear B, 48, the ratio of speed of gear A to gear $B = \frac{48}{60} = \frac{4}{5}$, or 4 to 5.

Compound Gearing

The simplest and most common case of *compound* gearing consists of four gears mounted as shown in Fig. 34. The gear A is the driver and meshes with gear B, which is keyed to the same shaft as gear C. Thus, when gear B is driven by gear A, gear C will revolve at the same rate of speed as gear B, and will drive gear D; the motion of gear A is thus transmitted to gear D. The two gears B and C which serve to transmit the motion are called *intermediate* gears. The gears A and C are the *driving* gears, and B and D the *driven* gears.

If it is required to find the revolutions of the driven gear D when the number of revolutions of gear A and the numbers of teeth in all the gears are known, the following rule is used:

* See MACHINERY'S Jig Sheet No. 6A.

The revolutions per minute of the driven gear equals the revolutions per minute of the driving gear, times a fraction, the numerator of which is made up of the product of the numbers of teeth in the driving gears, and the denominator of the product of the numbers of teeth in the driven gears.

This rule can be written as a formula as below:

$$\frac{\text{rev. per min.}}{\text{of driven gear}} = \frac{\text{rev. per min.}}{\text{of driving gear}} \times \frac{\text{product of teeth in driving gears}}{\text{product of teeth in driven gears}}$$

This formula can be used whether there be one or more sets of intermediate gears.

Assume that gear *A* in Fig. 34 has 40 teeth, gear *B*, 24 teeth, gear *C*, 50 teeth, and gear *D*, 25 teeth. Then if gear *A* makes 30 revolutions

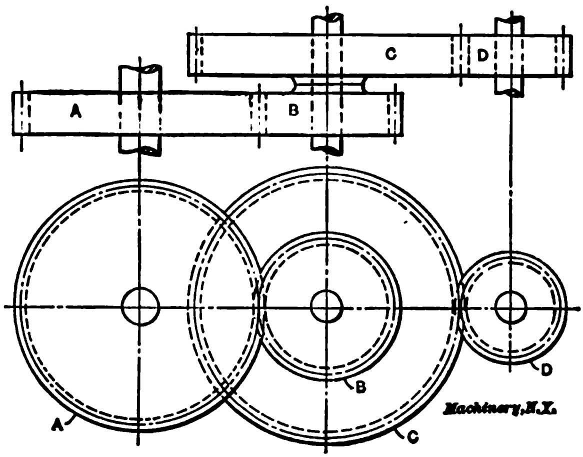


Fig. 34

per minute, how many revolutions does gear *D* make? Using the formula just given we have:

$$\text{rev. per min.} = 30 \times \frac{40 \times 50}{24 \times 25} = 100.$$

The revolutions per minute made by gear *B* may be found from the same formula by leaving out the numbers of teeth of gears *C* and *D*, thus:

$$\text{rev. per min.} = 30 \times \frac{40}{24} = 50.$$

Finding the Number of Teeth in Gears to Transmit Motion at a Given Ratio

If the numbers of revolutions of the driving gear *A* and the driven gear *D* are given, the numbers of teeth required in the four gears of a compound gearing that will transmit motion at the required ratio, can be found.

Assume that gear *A*, Fig. 34, makes 36 revolutions per minute and that it is required that gear *D* should make 56 revolutions. The speed ratio is then $\frac{36}{56} = \frac{9}{14}$. (See page 29.)

To find the gears required, write the ratio of the speed of the driving gear to the driven gear as a fraction, divide the numerator and denominator in two factors, and multiply each "pair" of factors by the same number until gears with suitable numbers of teeth are found. (One factor in the numerator and one in the denominator make "one pair.") In this example

$$\frac{9}{14} = \frac{3 \times 3}{2 \times 7} = \frac{(3 \times 20) \times (3 \times 10)}{(2 \times 20) \times (7 \times 10)} = \frac{60 \times 30}{40 \times 70}$$

The gears in the numerator, with 60 and 30 teeth, are the driven gears (gears *B* and *D*, Fig. 34), and the gears in the denominator, with 40 and 70 teeth, are the driving gears (gears *A* and *C*, Fig. 34).

The calculation may be expressed in a formula as follows:

$$\text{ratio of speed of the first driving gear to the last driven gear} = \frac{\text{product of teeth in driven gears}}{\text{product of teeth in driving gears}}$$

CHAPTER VI

LATHE CHANGE GEARING

While the principles and rules governing the calculation of change gears are very simple, they, of course, presuppose some fundamental knowledge of the use of common fractions. If such knowledge is at hand, the subject of figuring change gears, if once thoroughly understood, can hardly ever be forgotten. It should be impressed upon the minds of all who have found difficulties with this subject that the matter is seldom approached in a logical manner, and is usually grasped by the memory rather than by the intellect.

When cutting threads in the lathe, the lathe carriage is moved along the bed by means of the lead-screw a certain distance while the work revolves a certain number of times. If the work revolves 12 times while the carriage moves one inch along the bed of the lathe, 12 threads per inch will be cut on the work.

Change gears are used for transmitting the motion from the spindle (which revolves the work) to the lead-screw (which causes the carriage to move along the bed). The number of times that the spindle will revolve while the carriage moves one inch along the lathe bed is determined by the ratio of the change gears. By employing different ratios of change gearing, therefore, different numbers of threads per inch can be cut.

The change gearing may be either *simple* or *compound*. Simple gearing is shown in the accompanying illustration, Fig. 35. When simple gearing is used it is always necessary to use an idler between the gear on the spindle stud and the gear on the lead-screw. As already explained, this idler has no influence on the ratio of the gearing, and can have any number of teeth. Compound change gearing is shown in Fig. 36.

Finding the Lathe Screw Constant

In order to be able to calculate change gears for the lathe, it is necessary first to find the "lathe screw constant." This constant is always the same for each particular lathe, but it may be different for lathes of different sizes or makes.

To find the screw constant of a lathe, place gears with an equal number of teeth on the spindle stud and the lead-screw. Then cut a thread on a piece of work in the lathe. The number of threads per inch that

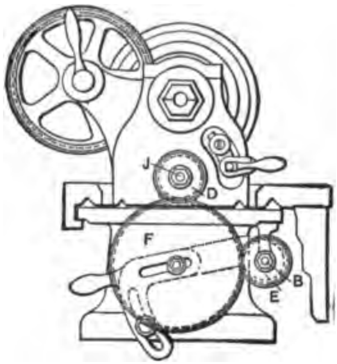


Fig. 35

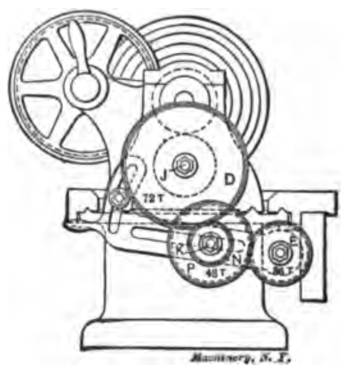


Fig. 36

will be cut on the work when gears with equal numbers of teeth are placed as directed is called the "screw constant" of the lathe.

For example, put gears with 48 teeth on the spindle stud and on the lead-screw, and any convenient gear on the intermediate stud. Then cut a thread on a piece between the centers. If the number of threads per inch when counted as shown on page 22 is found to be 8, the screw constant of this lathe is said to be 8.

To Find the Change Gears for Simple Gearing

When the lathe screw constant has been found, the number of teeth to be used in the change gears for cutting any number of threads within the capacity of the lathe can be determined as follows:

***Place the lathe screw constant as the numerator and the number of threads per inch to be cut as the denominator of a fraction and multiply the numerator and denominator by the same number until a new fraction results where the numerator and denominator represent suitable numbers of teeth for the change gears.* In the new fraction the numerator gives the number of teeth in the gear on the spindle stud, and the**

denominator the number of teeth in the gear on the lead-screw. This rule can be more easily remembered if written as a formula:

$$\frac{\text{lathe screw constant}}{\text{threads per inch to be cut}} = \frac{\text{teeth in gear on spindle stud}}{\text{teeth in gear on lead-screw}}$$

Assume that 10 threads per inch are to be cut in a lathe where we have found that the lathe screw constant is 6. Also assume that the numbers of teeth in the available change gears of this lathe are 24, 28, 32, 36, 40, etc., increasing by 4 up to 100. By substituting the figures given, in the formula above, and carrying out the calculation, we have:

$$\frac{6}{10} = \frac{6 \times 4}{10 \times 4} = \frac{24}{40}.$$

By multiplying both numerator and denominator by 4 we obtain two available gears with 24 and 40 teeth, respectively. The 24-tooth gear goes on the spindle stud, and the 40-tooth gear on the lead-screw. It will be seen that if we had multiplied 6 and 10 by 5, we would have obtained 30 and 50 teeth, which gears are not available in our set of gears with this lathe. Until getting accustomed to figuring of this kind, one can find, by trial only, the correct number by which to multiply the numerator and denominator. The trials must be kept up until both gears have such a number of teeth that they are to be found in the set of change gears accompanying the lathe.

Assume that it is required to cut 11½ threads per inch in the same lathe having the same set of change gears. Then

$$\frac{6}{11\frac{1}{2}} = \frac{6 \times 8}{11\frac{1}{2} \times 8} = \frac{48}{92}.$$

It will be found that multiplying by any other number than 8 would not, in this case, give numbers of teeth that could be found in the gears with the lathe. [In order to prevent mistakes, be sure to note that the lathe screw constant differs for different makes and sizes of lathes and must be determined for each particular lathe.]

Compound Gearing

Sometimes it is not possible to obtain gears that will give the required ratio for the thread to be cut in a simple train, and then compound gearing must be employed. The method for finding the number of teeth in the gears in compound gearing is exactly the same as for simple gearing, except that we divide both numerator and denominator of the fraction giving the ratio of screw constant to threads per inch to be cut, into two factors, and then multiply each "pair" of factors by the same number, in order to obtain the change gears. (One factor in the numerator and one in the denominator make one pair.)

Assume that the lathe screw constant is 6, that the numbers of teeth in the available gears are 30, 35, 40, 45, 50, 55, etc., increasing by 5 up to 100. Assume that it is required to cut 24 threads per inch. We have then,

$$\frac{6}{24} = \text{ratio}.$$

By dividing numerator and denominator of the ratio into two factors and multiplying each pair of factors by the same number, as shown below, we find the gears:

$$\frac{6}{24}=\frac{2\times 3}{4\times 6}=\frac{(2\times 20)\times(3\times 10)}{(4\times 20)\times(6\times 10)}=\frac{40\times 30}{80\times 60}.$$

The four numbers in the last fraction give the numbers of teeth in the gears which should be used. The gears in the numerator, with 40 and 30 teeth, are the driving gears, and those in the denominator, with 80 and 60 teeth, are the driven gears. Driving gears are, of course, the gear *D*, Fig. 36, on the spindle stud, and the gear *P* on the intermediate stud *K*, meshing with the lead-screw gear. Driven gears are the lead-screw gear, *E*, and the gear *N* on the intermediate stud meshing with the spindle stud gear. Either of the driving gears may be placed on the spindle stud, and either of the driven on the lead-screw.

Assume that 1¾ threads per inch are to be cut in a lathe with a screw constant 6, and that the gears available have 24, 28, 32, 36, 40 teeth, etc., increasing by 4 up to 100. Proceeding as before we have:

$$\frac{6}{1\frac{3}{4}}=\frac{2\times 3}{1\times 1\frac{3}{4}}=\frac{(2\times 36)\times(3\times 16)}{(1\times 36)\times(1\frac{3}{4}\times 16)}=\frac{72\times 48}{36\times 28}.$$

This is the case directly illustrated in Fig. 36. The gear with 72 teeth is placed on the spindle stud *J*, the one with 48 on the intermediate stud *K*, meshing with the lead-screw gear. These two gears (72- and 48-teeth) are the ***driving*** gears. The gears with 36 and 28 teeth are placed on the lead-screw, and on the intermediate stud, as shown, and are the ***driven*** gears.

The rule for compound change gears, given as a formula, is as follows:

$$\frac{\text{lathe screw constant}}{\text{threads per inch to be cut}}=\frac{\text{product of teeth in driving gears}}{\text{product of teeth in driven gears}}$$

Fractional Threads

Sometimes the lead of a thread is given as a fraction of an inch instead of stating the number of threads per inch. For instance, a thread may be required to be cut, having ⅜ inch lead. In this case the expression "⅜ inch lead" should first be transformed to "number of threads per inch," after which we can proceed to find the change gears as explained on the previous pages. How to find the number of threads per inch when the lead is given is explained in Chapter IV. The number of threads (the thread being single) equals:

$$\text{number of threads per inch}=\frac{1}{\frac{3}{8}}=1\div\frac{3}{8}=\frac{8}{3}=2\frac{2}{3}.$$

To find the change gears to cut 2 2/3 threads per inch in a lathe having a screw constant 8 and change gears running from 24 to 100 teeth, increasing by 4, proceed as below:

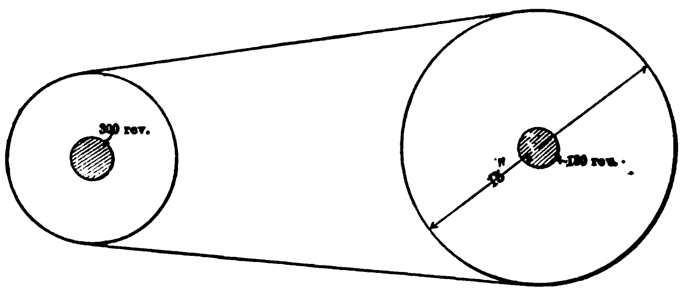
$$\frac{8}{2\,2/3}=\frac{2\times 4}{1\times 2\,2/3}=\frac{(2\times 36)\times(4\times 24)}{(1\times 36)\times(2\,2/3\times 24)}=\frac{72\times 96}{36\times 64}.$$

CHAPTER VII

SPEED OF PULLEYS

The principle applied to gearing in regard to the ratio between the speeds of two shafts, may be directly applied to the question of sizes of pulleys, with the only difference that we here deal with the number of inches to the diameter of the pulley instead of the number of teeth in the gear.

Assume that a shaft is required to make 300 revolutions per minute, and that it is driven from a line-shaft making 180 revolutions per minute, as shown in the illustration below. The pulley on the line-shaft is in place, and is 15 inches in diameter. What diameter should the pulley on the shaft making 300 revolutions per minute be made? As the belt on the two pulleys runs at the same speed as the periphery (circumference) of either of the pulleys, it is clear that the peri-



Machinery, N.Y.

Fig. 37

pheries of both pulleys run at the same speed, providing there is no slippage between the belt and the pulleys. The pulley running a smaller number of revolutions must, of course, be larger in order that its periphery may run at the same speed as the periphery of the pulley making a greater number of revolutions. The circumference of a circle (and, therefore, also the circumference of a pulley) equals the diameter $\times$ 3.1416. Therefore, the circumference of the pulley making 180 revolutions and having a diameter of 15 inches, passes, in one minute, through a distance equal to 180 times its circumference, or $180 \times 15 \times 3.1416$.

The circumference of the pulley making 300 revolutions must pass through the same distance in one minute; therefore, for *each* revolution this pulley must pass through the distance $180 \times 15 \times 3.1416$ divided by 300. This, then, would equal the circumference of the smaller pulley; but the circumference also equals the *diameter* $\times$ 3.1416. We can therefore write

$$\frac{180 \times 15 \times 3.1416}{300} = \text{diameter of smaller pulley} \times 3.1416$$

As 3.1416 enters as a factor on both sides of the equals sign, it can be cancelled. Then we have:

$$\frac{180 \times 15}{300} = \text{diameter of smaller pulley.}$$

From this we can formulate the following rule for the relation between the sizes of pulleys and the number of revolutions of two shafts:

The number of revolutions of one shaft multiplied by the diameter of the pulley on the same shaft divided by the number of revolutions of the second shaft, gives the diameter of the pulley on the second shaft.

We can write this rule in the form of a formula, as follows:

$$\begin{array}{c}\text{diameter of pulley on}\\ \text{second shaft in inches}\end{array} = \frac{\begin{array}{c}\text{number of revolutions}\\ \text{of first shaft}\end{array} \times \begin{array}{c}\text{diameter of pulley on}\\ \text{first shaft in inches}\end{array}}{\text{number of revolutions of second shaft}}$$

If one pulley makes 200 revolutions while another pulley makes 100 revolutions, we say that the *speed ratio* between the two pulleys is 2 to 1. If one pulley makes 200 revolutions while another makes 50 revolutions, the speed ratio is 4 to 1, because one shaft makes four times as many revolutions as the other. If one shaft runs 200 revolutions and another 140 revolutions, the speed ratio would be 200 to 140. By cancelling equal factors in 200 and 140, we can reduce the ratio so as to express it with smaller numbers. In this case the ratio would be 10 to 7. $\left(\frac{200}{140} = \frac{20}{14} = \frac{10}{7}\right)$

[If very accurate results are required, one thickness of the belt should be added to the diameter of the pulley itself, and the dimension thus obtained should be used in the formulas above instead of the diameter of the pulley rim. For instance, if the pulley is 5 inches in diameter and the belt 1/8 inch thick, the diameter to be used in the formulas should be 5 1/8 inches. The results obtained in this manner will be very accurate provided there is no slipping of the belt on either of the pulleys. For ordinary practical purposes, however, it is usual to figure with the diameter of the pulley rim, taking no account of the thickness of the belt.]

Finding the Number of Revolutions of a Shaft

If we know the diameters of the pulleys and the number of revolutions per minute of one shaft, and want to find the number of revolutions of the other shaft, we use the following rule:

The number of revolutions on one shaft multiplied by the diameter of the pulley on the same shaft, divided by the diameter of the pulley on the second shaft, gives the number of revolutions of the second shaft.

This rule can be written as a formula as follows:

$$\begin{array}{c}\text{number of revolutions}\\ \text{of second shaft}\end{array} = \frac{\begin{array}{c}\text{number of revolutions}\\ \text{of first shaft}\end{array} \times \begin{array}{c}\text{diameter of pulley}\\ \text{on first shaft}\end{array}}{\text{diameter of pulley on second shaft}}.$$

Practical Applications

Example 1. A line-shaft for some grinding machines is required to run at 320 revolutions per minute and is to be driven from a main line-shaft running at 200 revolutions per minute. The pulley on the main line shaft is already in place and is 24 inches in diameter. What diameter ought the pulley of the grinding machine line-shaft to be? If we apply our first rule or formula we have directly:

$$\frac{200 \times 24}{320} = 15$$

The diameter of the pulley on the grinding machine line-shaft therefore should be 15 inches.

Example 2. The pulley on a geared-head lathe is 16 inches in diameter. The driving pulley on the line-shaft is 34 inches in diameter, and the line-shaft makes 120 revolutions per minute. How many revolutions does the 16-inch pulley on the lathe make?

If we apply the second rule or formula given we have directly:

$$\frac{120 \times 34}{16} = 255$$

The 16-inch pulley consequently makes 255 revolutions per minute.

Example 3. The largest step of a cone-pulley on the countershaft of a machine is 12 inches in diameter. The smallest step is 6 inches. The largest step on the cone-pulley on the machine is 10 inches, the smallest, 4. If the countershaft runs at 300 revolutions per minute, what are the highest and lowest speeds of the spindle on which the cone-pulley on the machine is mounted?

The largest step of each respective pulley runs with the smallest step of the other. We can, therefore, proceed as if we had two sets of pulleys, one set 12 and 4, and one, 6 and 10 inches in diameter, and by using the second rule or formula given we have:

$$\frac{300 \times 12}{4} = 900 \quad \text{and} \quad \frac{300 \times 6}{10} = 180$$

The highest speed is therefore 900 revolutions per minute and the lowest 180 revolutions.

CHAPTER VIII

CHANGE GEARS FOR MILLING SPIRALS

The method for the figuring of change gears for cutting spirals on the milling machine, is, in principle, exactly the same as that used for figuring change gears for the lathe (see Chapter VI), but it will be necessary to shortly refer to the construction of the mechanism for connecting the index head spindle and the feed-screw to make perfectly clear the fundamental ideas governing the selection of change gears.

In Fig. 38 is shown an end view of an index head for a milling machine, placed on the top of the milling machine table. At *A* is

shown the end of the table feed-screw, and B is a gear placed on this feed-screw. This gear is commonly called the feed-screw gear, and it imparts motion, through an intermediate gear H, to the gear C which is placed on the stud D; from this stud, in turn, motion is imparted by gearing to the worm of the index head and from the worm to the worm-wheel mounted on the index head spindle. Thus, when connected by gearing in this manner, the index head spindle may be rotated from the feed-screw. The gear C on the stud D is called the "worm gear"; this worm gear should not be confused with the worm-wheel which is permanently attached to the index head spindle.

In Fig. 38 is shown a case of simple gearing, while in Fig. 39 the gears are compounded. In this case B still represents the feed-screw gear, while E is the gear on the intermediate stud which meshes with B, and F is the second gear on the same intermediate stud, meshing with gear C. The object of the calculation is to find the numbers of teeth in gears B and C used in a simple train, as in Fig. 38; or in the gears B, E, F and C as used in a compound train of gears, as shown in Fig. 39.

The Lead of a Milling Machine

If gears with an equal number of teeth are placed on the feed-screw A and the stud D in Fig. 38. then the *lead of the milling machine* is the distance the table will travel while the index spindle makes one complete revolution. This distance is a constant used in figuring the change gears, and may vary for different milling machines.

The lead of a helix or spiral is the distance, measured along the axis of the work, in which the spiral makes one full turn around the work. The lead of the milling machine may, therefore, also be expressed as the lead of the spiral that will be cut when gears with an equal number of teeth are placed on studs A and D, and an idler of suitable size interposed between the gears.

To find the lead of a milling machine, place equal gears on stud D, and on feed-screw A, Fig. 38, and multiply the number of revolutions made by the feed-screw to produce one revolution of the index head spindle, by the lead of the thread on the feed-screw.

We can express the rule given as a formula:

$$\begin{array}{c}\text{lead of milling}\\ \text{machine}\end{array} = \begin{array}{c}\text{rev. of feed-screw for one}\\ \text{revolution of index spindle}\\ \text{with equal gears}\end{array} \times \begin{array}{c}\text{lead of}\\ \text{feed-screw}\end{array}$$

Assume that it is necessary to make 40 revolutions of the feed-screw to turn the index head spindle one complete revolution, when the gears B and C, Fig. 38, are equal, and that the lead of the thread on the feed-screw of the milling machine is ¼ inch; then the lead of the machine equals

$$40 \times \tfrac{1}{4} \text{ inch} = 10 \text{ inches.}$$

Change Gears for Cutting Spirals

As has already been stated, the lead of the machine means the distance which the table of the milling machine moves forward in order to turn the work placed on the index head spindle one complete revo-

lution when change gears with an equal number of teeth are used. If then, for instance, a spiral is to be cut, the lead of which is twice as long as the lead of the machine, change gears of such a ratio must be used that the index head spindle will turn only one-half a revolution while the table moves forward a distance equal to the lead of the machine.

Assume that we want to cut a spiral having a lead of 20 inches, that is, making one complete turn in a length of 20 inches, and that the lead of the milling machine is 10 inches. Then the ratio between the

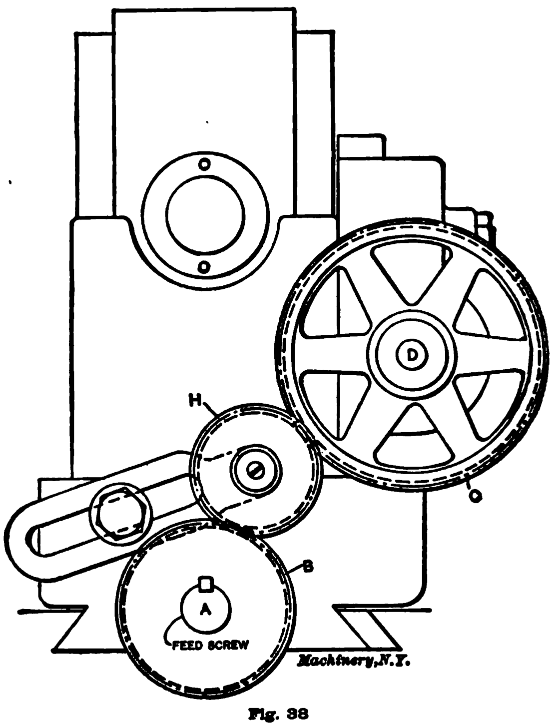


Fig. 38

speeds of the feed-screw and of stud *D* must be 2 to 1, which means that the feed-screw, which is required to turn twice while stud *D* turns once, must have a gear that has only one-half the number of teeth of the gear placed on stud *D*. (See Chapter V.) If the lead of the machine is 10 inches and the lead of the spiral required to be cut on a piece of work is 30 inches, then the ratio between the speed of the gears would be 3 to 1, which is the same as the ratio between the lead of the spiral to be cut to the lead of the machine. (30 to 10 = 3 to 1, or as it is commonly written 30 : 10 = 3 : 1.)

The rule for finding the change gears can be expressed by a simple formula:

$$\frac{\text{lead of spiral to be cut}}{\text{lead of milling machine}} = \frac{\text{number of teeth in gear on worm stud } (D, \text{ Fig. } 38)}{\text{number of teeth in gear on feed-screw}}$$

Expressed in words this formula would read:

To find the change gears to be used in a simple train of gearing when cutting spirals in the milling machine, place the lead of the spiral as the numerator and the lead of the milling machine as the denominator of a fraction, and multiply the numerator and denominator by

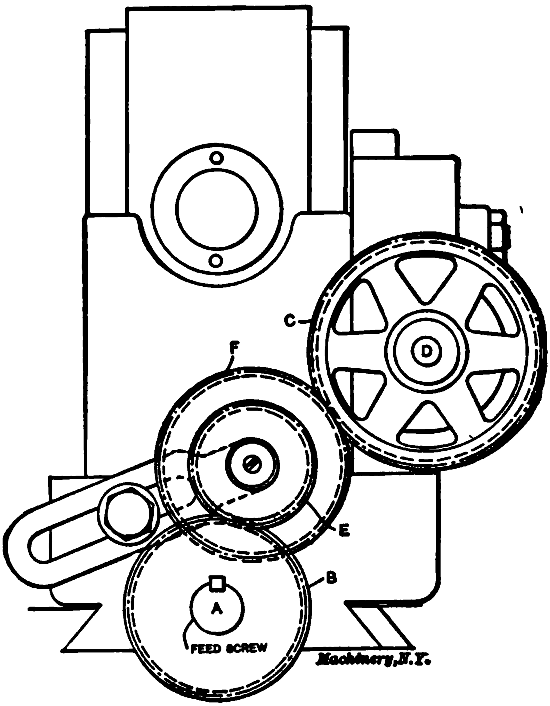


Fig. 39

the same number until a new fraction is obtained in which the numerator and denominator give suitable numbers of teeth for the gears.

As an example of this rule, take the case of a milling machine in which there are four threads per inch on the feed-screw and where 40 revolutions of the feed-screw are necessary to make the index spindle turn one complete revolution when gears *B* and *C*, Fig. 38, are equal. It is required on this machine to cut a spiral, the lead of which is 12 inches.

We first find the lead of the machine. As the feed-screw is single and has 4 threads per inch, the lead of the screw thread is ¼ inch (see Chapter IV), which multiplied by 40 is

$$40 \times \tfrac{1}{4} \text{ inch} = 10 \text{ inches} = \text{lead of machine.}$$

To find the change gears, place the lead of the required spiral as the numerator of a fraction and the lead of the machine as the denominator, and multiply both numerator and denominator by the same number until a new fraction is obtained in which the numerator and denominator express suitable numbers of teeth. Following this rule, then,

$$\frac{12}{10} = \frac{12 \times 4}{10 \times 4} = \frac{48}{40}$$

The gear with 48 teeth is placed on stud D which, of course, is required to revolve slower than the lead-screw, in order to cut a spiral which is 12 inches, when the spiral cut with equal gears is only 10 inches. The gear having 40 teeth is placed on the feed-screw. An intermediate gear is put between the gear on the feed-screw and the gear on stud D; the number of teeth in this intermediate gear has no influence on the speed ratio of feed-screw A and stud D, but simply serves to transmit motion from one gear to the other. (See Chapter V.)

Gears for Compound Gearing

If it is not possible to find a set of two gears that will transmit the required motion, it is necessary to use compound gearing. In this case the manner in which the gears are found is exactly the same as the method used for compound gearing in the lathe. The lead of the required spiral is placed as the numerator and the lead of the milling machine as the denominator of a fraction. Then the numerator and denominator are divided into two factors each, and then each "pair" of factors (one factor in the numerator and one in the denominator making one "pair") is multiplied by the same number until suitable numbers of teeth for the change gears are obtained.

As an example, assume that the lead of a machine is 10 inches, and that a spiral having a 48-inch lead is to be cut. Following the method explained we then have.

$$\frac{48}{10} = \frac{6 \times 8}{2 \times 5} = \frac{(6 \times 12) \times (8 \times 8)}{(2 \times 12) \times (5 \times 8)} = \frac{72 \times 64}{24 \times 40}$$

The gear having 72 teeth is placed on the stud D and meshes with the 24-tooth gear F (see Fig. 39) on the intermediate stud. On the same intermediate stud is then placed the gear having 64 teeth, which is driven by the gear having 40 teeth placed on the feed-screw. This makes the gears having 72 and 64 teeth the driven gears, and the gears having 24 and 40 teeth the driving gears, the whole train of gears being driven from the feed-screw of the table.

In general, for compound gearing, the following formula may be used:

$$\frac{\text{lead of spiral to be cut}}{\text{lead of machine}} = \frac{\text{product of driven gears}}{\text{product of driving gears}}$$

CHAPTER IX

MILLING MACHINE INDEXING

The figuring of indexing movements for the dividing head of the milling machine is a subject which many mechanics think complicated, although it is really very simple. Assume that a bolt having a round head as shown in Fig. 40 is required to be milled so that the head becomes hexagonal, that is, so that it has six equal sides, as shown in Fig. 41. The index head is used for holding the work and for turning or indexing it the required amount for milling each of the six flat surfaces in turn. The index head is constructed with a worm and worm-wheel mechanism, the worm being on the crank turned when indexing, and the worm-wheel being mounted on the index spindle to which the work is attached. By moving the crank with its index pin a certain number of holes in one of the index circles, a certain angular movement can be imparted to the work. The calculating of indexing movements for the milling machine consists in finding how much the

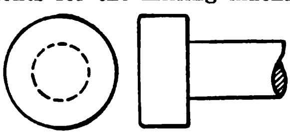


Fig. 40

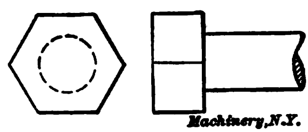


Fig. 41

index crank requires to be turned in order to produce the required movement for indexing the work.

Calculating the Indexing Movement

Most of the regularly manufactured index heads use a single threaded worm engaging with a worm-wheel having 40 teeth. Thus, when the index crank is turned around one full revolution, the worm is also revolved one complete turn, and this moves the worm-wheel one tooth, or 1/40 of its circumference. Therefore, in order to turn the worm-wheel and the index spindle on which it is mounted one full revolution, it is necessary to turn the index crank 40 revolutions. If we want to revolve the index spindle one-half revolution, we would have to turn the index crank 20 revolutions. If we want to turn the index spindle only one-fourth of a revolution, we turn the index crank 10 revolutions.

Suppose that we want to mill the hexagonal head of the bolt shown in Fig. 41. As it requires 40 revolutions of the index crank to revolve the index spindle once, it evidently requires only 1/6 of that number to turn the index spindle 1/6 revolution; this is the amount that the work should be turned around or indexed when one side of the hexagon has been milled, and we are ready to mill the next. Consequently, the index crank should be turned around $\frac{40}{6} = 6\ 2/3$ revolutions for milling

a hexagon; that is, we first turn the crank 6 full revolutions and then by means of the index plate we turn it 2/3 of a revolution. If we use the circle in the index plate having 18 holes, 2/3 of a revolution will mean 12 holes in this circle, as 12 is two-thirds of 18 ($12 = 2/3 \times 18$).

Assume that a piece of work has eight sides regularly spaced (regular octagon). The indexing for each side is found by dividing 40 by 8. Thus $\frac{40}{8} = 5$, represents the number of revolutions of the index crank for each side indexed and milled.

Assume that it is required to cut nine flutes regularly spaced in a reamer. The index crank must be turned $\frac{40}{9} = 4\,4/9$ revolutions in order to index for each flute. The 4/9 of a revolution would correspond to eight holes in the 18-hole circle, because $\frac{8}{18} = \frac{4}{9}$.

Assume that it is required to cut 85 teeth in a spur gear. The index crank must be revolved $\frac{40}{85} = \frac{8}{17}$ revolutions to index for each tooth. To move the index crank $\frac{8}{17}$ of a revolution corresponds to moving it 8 holes in the 17-hole circle.

As a general rule, for finding the number of revolutions required for indexing for any regular spacing, with any index head, the following rule may be used: *To find the number of revolutions of the index crank for indexing, divide the number of turns required of the index crank for one revolution of the index head spindle by the number of divisions required in the work.*

[Most standard index heads are provided with an index plate fastened directly to the index spindle for rapid direct indexing. This index plate is usually provided with 24 holes, so that 2, 3, 4, 6, 8, 12 and 24 divisions can be obtained directly by the use of this direct index plate without using the regular indexing mechanism. When using this index plate for rapid direct indexing, no calculations are required, as the number of divisions obtainable by the use of the different holes in this plate are, as a rule, marked directly at the respective holes.]

Finding the Index Circle to Use

In order to find which index circle to use and how many holes in that index circle to move for a certain fractional turn of the index crank, the numerator and denominator of the fraction expressing the fractional turn are multiplied by the same number until the denominator of the new fraction equals the number of holes in some one index circle. The number with which to multiply must be found by trial. The numerator of the new fraction then expresses how many holes the crank is to be moved in the circle expressed by the denominator.

Assume that 12 flutes are to be milled in a large tap. Assume that 40 turns of the index crank are required for one turn of the index

head spindle. First divide 40 by 12 to find the number of turns of the index crank required for each indexing. Writing out this division as a fraction and carrying out the calculation gives us

$$\frac{40}{12} = 3\ 4/12 = 3\ 1/3.$$

The fractional turn required is 1/3 of a revolution. Now multiply, according to the rule given, the numerator and denominator of this fraction by a number so selected that the new denominator equals the number of holes in some one index circle. Multiplying by 6 would give us

$$\frac{1 \times 6}{3 \times 6} = \frac{6}{18}$$

in which fraction 18 expresses the number of holes in the index circle to use, and 6 is the number of holes the crank must be moved in this circle to turn the worm shaft and worm one-third of a revolution.

Most milling machines are commonly furnished with three index plates, each having six index circles. The following numbers of holes in the index circles of the three index plates are commonly used:

15	16	17	18	19	20
21	23	27	29	31	33
37	39	41	43	47	49

CHAPTER X

INDEXING FOR ANGLES

When two lines meet as shown in Fig. 42, they form an angle with each other. The point where the two lines meet or intersect is called the *vertex* of the angle. The two lines forming the angle are called the sides of the angle.

Angles are measured in degrees and subdivisions of a degree. If the circumference (periphery) of a circle is divided into 360 parts, each part is called one degree, and the angle between two lines from the center to the ends of the small part of the circle is a one-degree angle, as shown in Fig. 43. As the whole circle contains 360 degrees, one-half of a circle contains 180 degrees, and one-quarter of a circle, 90 degrees.

A 90-degree angle is called a *right* angle. An angle larger than 90 degrees is called an *obtuse* angle, and an angle less than 90 degrees is called an *acute* angle. (See Figs. 46, 47, and 48.) Any angle which is not a right angle is called an *oblique* angle.

When two lines form a right or 90-degree angle with each other, as shown in Fig. 44, one line is said to be *perpendicular* to the other.

Angles are said to be equal when they contain the same number of degrees. The angle in Fig. 49 and the angle in Fig. 50 are equal, be-

cause they are both 60 degrees; that the sides of the angle in Fig. 50 are longer than the sides of the angle in Fig. 49 has no influence on the angle because of the fact that an angle is only the *difference in direction* of two lines. The angle in Fig. 52 which contains only 30 degrees is only one-half of the angle in Fig. 49.

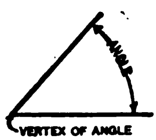
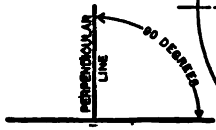


Fig. 42



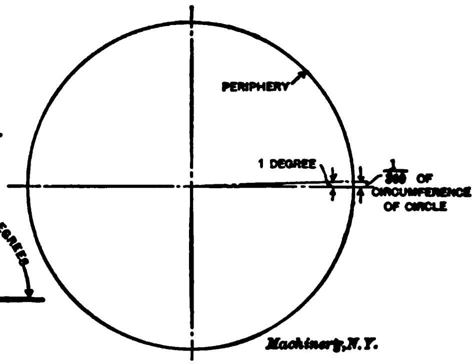


Fig. 44 Fig. 43

One-half of a right angle is 45 degrees, as shown in Fig. 51. In Fig. 53 is shown an angle which is 120 degrees, and which can be divided into a right or 90-degree angle, and a 30-degree angle.

In order to obtain finer subdivisions for the measurement of angles than the degree, one degree is divided into 60 minutes, and one minute into 60 seconds.

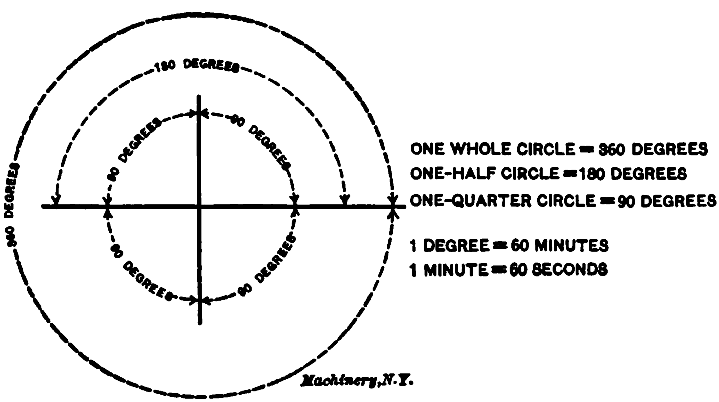


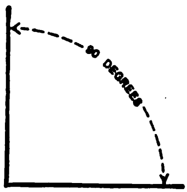
Fig. 45

Any part of a degree can be expressed in minutes and seconds, for instance, 1/2 degree = 30 minutes, 1/3 of a degree = 20 minutes; and since 1/4 of a degree = 15 minutes, 3/4 of a degree = 45 minutes. In the same way 1/2 minute = 30 seconds, 1/3 minute = 20 seconds, 1/4 minute = 15 seconds, and 3/4 minute = 45 seconds.

The word degree is often abbreviated "deg.," or the sign (°) is used to indicate degrees; thus 60° = 60 degrees. In the same way 60′ = 60 minutes (min.), 60″ = 60 seconds (sec.).

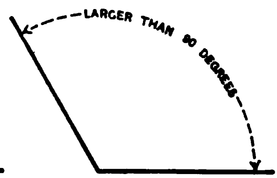
Indexing for Angles

In Fig. 54 is shown a piece of round stock having two flats milled in such a way that the angle between two lines from the center at right angles to the two surfaces is 35 degrees. In this case the index head cannot be turned so as to make a certain whole number of moves in one complete revolution of the work, as is done, for instance, when we make four moves in one revolution for milling a square, six moves



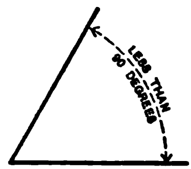
RIGHT ANGLE

Fig. 46



OBTUSE ANGLE

Fig. 47



ACUTE ANGLE

Machinery,N.Y.

Fig. 48

in one revolution for milling a hexagon, and 80 moves for milling an 80-tooth gear. Instead, we have here given a certain number of degrees which it is required that the work be turned before another cut is taken by the milling cutter.

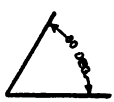


Fig. 49

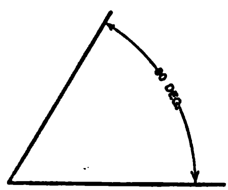


Fig. 50

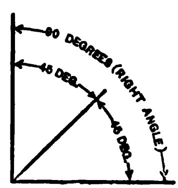


Fig. 51

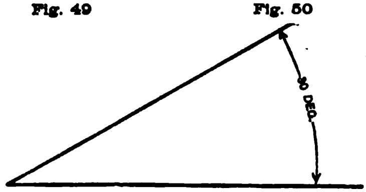
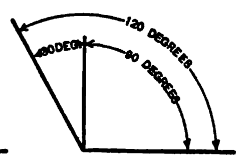


Fig. 52



Machinery,N.Y.

Fig. 53

Indexing for angles is only required whenever an angle is given which is not such a simple fraction of the whole circle as, for instance, 90 degrees, which is 1/4 of a complete turn, or 45 degrees, which is 1/8 of a complete turn, or 60 degrees, which is 1/6 of a complete turn; the numbers of turns of the index crank in these cases are determined as explained in Chapter IX. But if it be required to index for, say, 19 degrees, the method used is as explained in the following.

Calculating the Movements for Angular Indexing

There are 360 degrees in one complete circle or turn, and assuming that 40 turns of the index crank are required for one revolution of the

work, one turn of the index crank must equal $\frac{360}{40} = 9$ degrees. Then, when one complete turn of the index crank equals 9 degrees, two holes in the 18-hole circle, or 3 holes in the 27-hole circle, must correspond to one degree. $(\frac{3}{27} = \frac{2}{18} = \frac{1}{9}.)$ The first principle or rule for indexing for angles is therefore that two holes in the 18-hole circle or 3 holes in the 27-hole circle equals a movement of one degree of the index head spindle and the work.

Assume that an indexing movement of 35 degrees is required as shown in Fig. 54. One complete turn of the index crank equals 9 degrees; we, therefore, first divide the number of degrees for which we wish to index, by 9, in order to find how many complete turns the

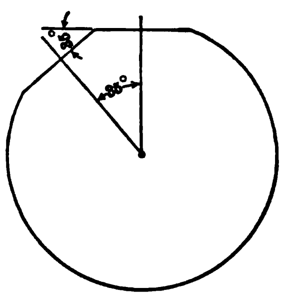
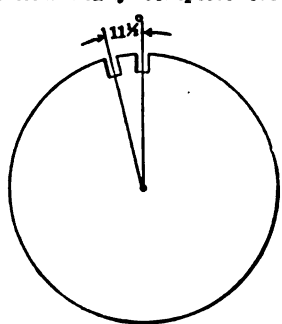


Fig. 54



Machinery,N.Y.

Fig. 55

index crank should make. The number of degrees left to turn when we have completed the full turns are indexed by taking two holes in the 18-hole circle for each degree. In the case in Fig. 54, $\frac{35}{9} = 3\frac{8}{9}$, which indicates that the index crank must be turned three revolutions, and then we must index for 8 degrees more or move 16 holes in the 18-hole circle.

Assume that we wish to index 11½ degrees, as shown in Fig. 55. Two holes in the 18-hole circle represent one degree, and consequently one hole represents ½ degree. To index for 11½ degrees we first turn the index crank one revolution, this being a 9-degree movement. Then to index 2½ degrees we must move the index crank 5 holes in the 18-hole circle (4 holes for the two whole degrees and one hole for the ½ degree equals the total movement of 5 holes).

Below is shown how this calculation may be carried out to plainly indicate the motion required for this angle:

11½ deg. = 9 deg. + 2 deg. + ½ deg.

1 turn + 4 holes + 1 hole in the 18-hole circle.

Should it be required to index only 1/3 degree, this may be made by using the 27-hole circle. In this circle a three-hole movement equals one degree, and a one-hole movement in that circle thus equals 1/3

degree, or 20 minutes. Assume that it is required to index the work through an angle of 48 degrees 40 minutes. First turn the crank 5 turns for 45 degrees ($5 \times 9 = 45$). Then there are 3 degrees 40 minutes or 3 2/3 degrees left. In the 27-hole circle a three-degree movement corresponds to 9 holes, and a 2/3-degree movement to 2 holes, making a total movement of 11 holes in the 27-hole circle, to complete the crank movement for 48 degrees 40 minutes. Below is plainly shown how this calculation may be carried out:

48 deg. 40 min. = 45 deg. + 3 deg. + 40 min.

5 turns + 9 holes + 2 holes in the 27-hole circle.

Approximate Indexing for Angles

By using the 18- and 27-hole circles, only whole degrees and 1/3, 1/2, and 2/3 of a degree (20, 30, and 40 minutes) can be indexed. Assume, however, that it is required to index for 16 minutes. One whole turn of the index crank equals 9 degrees or 540 minutes ($9 \times 60 = 540$). To index for 16 minutes, therefore, requires about 1/34 of a turn of the index crank ($540 \div 16 = 34$, nearly). In this case, therefore, we use an index circle having the nearest number of holes to 34, or the index circle with 33 holes. A one-hole movement in this circle would approximate the required movement of 16 minutes.

Assume that it is required to index for 55 minutes. We then have $540 \div 55 = 10$, nearly. In this case there is no index circle with 10 or approximately 10 holes, but as there is an index circle with 20 holes, this circle will be used, and the index crank is moved two holes in that circle instead of one.

Assume that it is required to index for 2 degrees 46 minutes. If we change this to minutes we have 2 degrees $= 2 \times 60 = 120$ minutes, and 46 minutes added to this gives us a total of 166 minutes. Dividing 540 by 166 we have: $540 \div 166 = 3.253$.

Now we multiply this quotient (3.253) by some whole number, so that we obtain a product which equals the number of holes in any one index circle. The number by which to multiply must be found by trial. In this case we can multiply by 12, obtaining as a product $3.253 \times 12 = 39.036$. For indexing 2 degrees and 46 minutes we can, therefore, use the 39-hole circle, moving the index crank 12 holes.

The following is a general rule for *approximate* indexing of angles, for any index head where 40 revolutions of the index crank are required for one revolution of the work:

***Divide* 540 *by the total number of minutes to be indexed. If the quotient is approximately equal to the number of holes in any index circle, the angular movement is obtained by moving one hole in this index circle. If the quotient does not approximately equal the number of holes in any index circle, find by trial a number by which the quotient can be multiplied so that the product equals the number of holes in an available index circle; in this circle, move the index crank as many holes as indicated by the number by which the quotient has been multiplied.* (If the quotient of 540 divided by the total number of minutes is greater than the number of holes in any of the index circles, the movement cannot be obtained by simple indexing.)**

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NUMBER 19

USE OF FORMULAS IN MECHANICS

FOURTH REVISED EDITION

CONTENTS

NEW YORK
THE INDUSTRIAL PRESS
1917

Students whose knowledge of elementary arithmetic and its application to simple problems is too limited for intelligent study of this treatise, are advised to first study MACHINERY's Jig Sheets 5A to 15A, inclusive, Common Fractions and Decimals; MACHINERY's Reference Series No. 18, Shop Arithmetic for the Machinist; and No. 52, Advanced Shop Arithmetic for the Machinist.

In preparing the second edition of this book, the chapter on graphical methods of solving problems, contained in the first edition, was omitted, and in its place a chapter containing solutions of twenty-four mechanical problems selected from many different fields of mechanical engineering, were introduced. This substitution, it is believed, greatly enhanced the value of the book, and met with the approval of readers especially interested in the use of formulas in mechanics. In the present—the fourth—edition, this feature has, therefore, been retained.

CHAPTER I

GENERAL REMARKS ON SELF-EDUCATION AND THE USE OF FORMULAS*

There are several ways of obtaining an education: The easiest and, until recent years, the usual way is to begin at the age of seven and continue steadily at school till the age of twenty-four, at father's expense. It is a fortunate fact that education is by no means unattainable otherwise; indeed many of the greatest and most widely useful educations the world has known have been obtained almost without a look at the inside of a school. A second method, quite modern, is the correspondence school—most excellent in many respects, yet not completing the available ways of obtaining an education. The final method is that of self-education. Nearly every successful man in engineering must necessarily obtain a very large share of his education in this manner, no matter what his general educational facilities have been; and it is for the purpose of explaining the possibilities of this method, and to plant the seed of self-help, that this and the following chapters have been written. They are divided into five heads dealing with the following subjects:

1. Present introduction, explaining general methods to be followed, and the principles of the use of formulas.
2. Examples of the use of formulas in mechanics.
3. The application of formulas to the solution of problems involving the principles of levers and moments, showing the simplicity of the form and application of the formulas.
4. The application of formulas in finding the center of gravity of geometrical figures.
5. The elements of the theory of the strength of materials, and the use of formulas in calculations of strength of beams.

It is the aim of these chapters to start the ambitious young man of sufficient grit upon a path which, if rightly followed, will in the future surely place him on par with those more fortunate men of his age who have enjoyed a college education, and to leave him in a position to continue to read and study and to understand the technical discussion and articles on design which appear in the technical press.

Engineering education does not consist in knowing things mechanical—far from it. It consists largely in knowing where to find technical literature upon any given subject when it is wanted, and knowing how to read it when it is found. Therefore, the first thing needed by our student is a place to store his newly acquired knowledge, aside from his head. The first attempt in this line of the author of this chapter, was a book having black canvas covers and a flexible back. Tapes were provided to lace in the leaves, which were made of fairly

* MACHINERY, October, 1905.

heavy cardboard, perforated for the tapes, and having a flexible strip along the perforated edge to enable the leaves to turn back properly. Twenty-six alphabet leaves were made similar to those in dictionaries and memorandum books, and a supply of extra leaves kept on hand.

Clippings from papers and catalogues were pasted on blank leaves and inserted under the proper letter, also notes and formulas received from others were written in, making the book a record of past work and study. The book, finally becoming too large to be convenient and too small to hold everything to be preserved, gave way to the card index and filing case.*

Having provided a systematic way to file our clippings, we are ready to consider the sources of the same. First subscribe for one or two of the leading technical journals devoted to your line of work. Make a practice of sending for catalogues of machinery manufacturers, and file them in the filing box. Many catalogues present, besides the goods manufactured, tables and data of value. If you can clip out these tables and file them in the card index without destroying the catalogue, do so; if not, make an entry in the card index to show where they may be found, before filing the catalogue. Always write your name in the catalogues, for as the file grows, you will find demands upon it from others, and this will aid in keeping the file intact. Remember that a catalogue received implies confidence on the part of the sender that it will eventually prove of use to him by bringing his goods before possible purchasers, and for this reason, as well as for your own convenience, all catalogues received should be listed and filed.

Duplicate clippings, such as tables, may often be exchanged with others, and thus our files are enlarged. This is not meant to encourage a mere mania for collecting—far from it. We should so study all data filed as to understand it at the time, and if found difficult, make such notes as will readily recall the study to our minds in the future.

Mathematical Signs and Expressions

The first thing to be done in preparation for study, and for reading the technical papers, is to become familiar with the *engineering language*. The *spoken engineering language* is of course the native tongue of the country, with, however, plenty of new words to master; but the *written engineering language* consists very largely of symbols, so like those of higher mathematics in appearance as often to discourage the beginner from further efforts. In the *written engineering language*, rules, instead of being written in the native tongue, are expressed by combinations of these symbols, and when so expressed are called formulas.

Now, the mathematician, when deriving a formula, uses the same symbols as the engineer when writing a formula, and if we accept the work of the mathematician as correct, we need pay no attention to the use of these symbols in deriving formulas, but give our attention to learning to read the symbolic language of the engineer with sufficient

* See MACHINERY'S Reference Series No. 2, Drafting-Room Practice, second edition, page 44: Card Index for the Draftsman's Individual Records.

ease to enable us to follow the operations called for by any formula we may wish to use.

The following table exhibits in the first column the symbols most frequently met with; in the second column the arithmetical equivalent of the symbols is given, assuming that $a = 2$ and $b = 4$; in the third column the symbols are expressed in English to give the proper method of reading the symbols.

TABLE 1. COMMON MATHEMATICAL SIGNS

$a = 2 \quad b = 4$		a equals 2 $\quad b$ equals 4
$a + b = c$	$2 + 4 = 6$	a plus b equals c
$b - a = d$	$4 - 2 = 2$	b minus a equals d
$a \times b = e$, $a.b = e$, $a\,b = e$	$2 \times 4 = 8$	a times b equals e, or ab equals e
$a\,(a + b) = f$	$2 \times 6 = 12$	a times a plus b equals f
$b \div a = h$, $\dfrac{b}{a} = h$	$\dfrac{4}{2} = 2$	b divided by a equals h, or b over a equals h
$a < b$	$2 < 4$	a is less than b
$b > a$	$4 > 2$	b is greater than a
$b:a::f:c$, $\dfrac{b}{a} = \dfrac{f}{c}$	$\dfrac{4}{2} = \dfrac{12}{6}$	b is to a as f is to c; b divided by a equals f divided by c; b over a equals f over c
$a^2 = b$	$2 \times 2 = 4$	a square equals b*
$b^3 = k$	$4 \times 4 \times 4 = 64$	b cube equals k
$\sqrt{b} = a$	$\sqrt{4} = 2$	square root of b equals a
$\sqrt[3]{e} = a$	$\sqrt[3]{8} = 2$	cube root of e equals a

Examples of Formulas

Let us now take the simple case of finding the area of a circle whose diameter we know. Expressed in English the rule is: Multiply the diameter by itself, then multiply the resulting product by 0.7854. The result is the area of the circle. If the diameter is expressed in inches, the area will be expressed in square inches. The corresponding mathematical expression is

$$A = 0.7854\, d^2 \tag{1}$$

where $A =$ the area in square inches,

$d =$ the diameter in inches.

Note that d^2 simply means $d \times d$.

Now, to solve this expression for a particular case, suppose we wish to know the area of a circle nine inches in diameter. We simply substitute for d^2 its numerical value, and perform the indicated operation, thus:

$$A = 0.7854 \times 9 \times 9 = 0.7854 \times 81 = 63.617 \text{ square inches.}$$

* For a more complete explanation of the meaning of square and square root, and cube and cube root, see MACHINERY'S Reference Series No. 52, Advanced Shop Arithmetic for the Machinist, or MACHINERY'S Jig Sheets No. 19A, Square Root, and No. 20A. Cube Root.

Take as another example the formula for the indicated horse-power of an engine:

$$H.P. = \frac{PLAN}{33{,}000} \qquad (2)$$

where P = the mean effective pressure in pounds per square inch,
L = the length of stroke in feet,
A = the area of the piston in square inches,
N = the number of strokes per minute.

Note that $PLAN$ simply means $P \times L \times A \times N$.*

The whole information as to how to determine the indicated horse-power of an engine is given in a very small space in the formula, while to write the same in English would require considerable of the space at our disposal.

Take the case of an 8×10-inch engine running at 100 revolutions per minute under 125 pounds mean effective pressure; here we have:

$P = 125$ pounds,

$$L = \frac{10 \text{ inches}}{12} = 0.833 \text{ feet,}$$

$A = 0.7854 \times 8 \times 8 = 50.26$ square inches,

$N = 100$ rev. per min. $\times 2 = 200$.

Then,

$$H.P. = \frac{125 \times 0.833 \times 50.26 \times 200}{33{,}000} = 31.7$$

Right-angled Triangles

In right-angled triangles,† if we call the side opposite the right angle a, and the sides forming the right angle b and c, then the following formula expresses the relationship between the three sides:

$$a = \sqrt{b^2 + c^2} \qquad (3)$$

Assume, for example, that in a right-angled triangle one of the sides forming the right angle is 8 inches long, and the other side forming the right angle is 6 inches. What is the length of the side opposite the right angle?

If we insert the given dimensions in the formula above, we have:

$$a = \sqrt{8^2 + 6^2} = \sqrt{64 + 36} = \sqrt{100} = 10.$$

The side opposite the right angle, thus, is 10 inches long.

* See MACHINERY'S Reference Series No. 52, Advanced Shop Arithmetic for the Machinist, or MACHINERY'S Jig Sheet No. 16A, Use of Formulas.

† See MACHINERY'S Jig Sheet No. 21A, Squares, Rectangles, Triangles, etc. For a more complete treatment of the right-angled triangle see MACHINERY'S Reference Series No. 52, Advanced Shop Arithmetic for the Machinist, and No. 54, Solution of Triangles.

CHAPTER II

THE USE OF FORMULAS IN MECHANICS

The use of formulas for solving problems in mechanics can best be made clear by actual examples. In the present chapter, therefore, a number of problems have been solved, showing the methods employed, and the manner in which the formulas taken from hand books and reference works are used.

Problem 1.—A metal ball falls from the top of a tower 300 feet high. How long a time will be required before it reaches the ground?

The formula by means of which this problem is solved is:*

$$t = \sqrt{\frac{2h}{g}} \tag{4}$$

in which t = time in seconds,

h = height in feet,

g = acceleration due to gravity = 32.16 feet.

Inserting the known values of h and g in the formula, we have:

$$t = \sqrt{\frac{2 \times 300}{32.16}} = \sqrt{18.66} = 4.32 \text{ seconds.}$$

Problem 2.—What is the velocity of the ball in the previous example when it reaches the ground?

The formula for finding the velocity is:

$$v = \sqrt{2gh} \tag{5}$$

in which v = velocity in feet per second, and h and g denote the same quantities as in Problem 1. Inserting the values of g and h in the formula, we have:

$$v = \sqrt{2 \times 32.16 \times 300} = \sqrt{19{,}296} = 139 \text{ feet, nearly.}$$

Problem 3.—A projectile is fired from a 12-inch gun vertically into the air. It strikes the ground, coming down, exactly 1 minute and 40 seconds after it left the muzzle. Disregarding air resistance, what height did the projectile reach? What was its velocity when leaving the muzzle? And what is the energy of the projectile when it strikes the ground, if its weight is assumed to be 600 pounds?

The time required for the projectile to reach its greatest height is one-half of the total time for the upward and downward journey. Thus, in 50 seconds, the projectile has reached the point where its velocity is zero, and where it begins to fall. The formula for finding the height reached is:

$$h = \frac{gt^2}{2} \tag{6}$$

* See MACHINERY'S Reference Series No. 5, First Principles of Theoretical Mechanics, page 34, second edition.

in which h, g and t denote the same quantities as in Problem 1. Inserting the known values, we have:

$$h=\frac{32.16\times 50^2}{2}=\frac{32.16\times 2{,}500}{2}=40{,}200 \text{ feet,}$$

$$\text{or } \frac{40{,}200}{5{,}280}=7.6 \text{ miles, approximately.}$$

The velocity of the projectile when leaving the muzzle is the same as the velocity acquired when again reaching the ground. This velocity is found by the formula:

$$v=gt=32.16\times 50=1{,}608 \text{ feet per second.} \qquad (7)$$

The energy of the projectile when it strikes the ground equals its weight multiplied by the distance through which it has fallen. If $W=$ weight, and $E=$ energy, we have:

$$E=W\times h=600\times 40{,}200=24{,}120{,}000 \text{ foot-pounds.} \qquad (8)$$

Another formula for the energy is as follows:

$$E=\frac{Wv^2}{2g}. \qquad (9)$$

This formula, with the values of W, v and g inserted, will, of course, give the same result.

$$E=\frac{600\times 1{,}608^2}{2\times 32.16}=\frac{600\times 2{,}585{,}664}{2\times 32.16}=24{,}120{,}000 \text{ foot-pounds.}$$

If, upon reaching the ground, the projectile buries itself to a depth of 8 feet, what is the average force of the blow with which it strikes the ground? The average force of the blow equals the energy divided by the distance d in which it is used up, plus the weight of the projectile, or if $F=$ average force of blow:

$$F=\frac{E}{d}+W=\frac{24{,}120{,}000}{8}+600=3{,}015{,}600 \text{ pounds.} \qquad (10)$$

Problem 4.—A drop hammer weighing 300 pounds falls through a distance of 3 feet. What is the stored or kinetic energy of the hammer when it strikes the work, and what is the average force with which it delivers the blow, if, on striking the work, it compresses it ½ inch?

From Formula (8) given in Problem 3, we have:

$$E=W\times h=300\times 3=900 \text{ foot-pounds.}$$

The distance d in which this energy is used up equals ½ inch or $\frac{1}{2}\div 12=0.04$ foot. Therefore, from Formula (10) the average force is:

$$F=\frac{E}{d}+W=\frac{900}{0.04}+300=22{,}500+300=22{,}800 \text{ pounds.}$$

Problem 5.—Find the stress in the rim of a fly-wheel, 5 feet mean diameter, made of cast iron, the rim being 2 inches wide by 4 inches deep, if the fly-wheel rotates at a velocity of 200 revolutions per minute.

The formula for the stress in the rim is:*

$$S = 0.00005427\, WRr^2 \qquad (11)$$

in which S = stress in pounds on the rim section,
W = weight of rim in pounds,
R = mean radius in feet, and
r = revolutions per minute.

We know that the mean diameter of the fly-wheel is 5 feet; therefore, $R = 2.5$ feet; r is given as 200; but we must find the value of W before we can apply Formula (11).

The weight W of the rim equals its volume or content in cubic inches multiplied by the weight of cast iron per one cubic inch. The volume of the rim equals the cross-sectional area of the rim multiplied by the circumference of the circle having for radius the mean radius of the flywheel; expressed as a formula:

$$V = 2\,R \times 3.1416 \times a \times b.$$

in which V = the volume of the rim, in cubic inches, R = the mean radius, in inches, a = the width, and b = the depth of the rim, in inches. Substituting the values in this formula, we have:

$$V = 2 \times 30 \times 3.1416 \times 2 \times 4 = 1{,}508 \text{ cubic inches.}$$

One cubic inch of cast iron weighs 0.26 pound. The weight of the rim then is:

$$W = 1{,}508 \times 0.26 = 392 \text{ pounds.}$$

We can now substitute the values in Formula (11):

$$S = 0.00005427 \times 392 \times 2.5 \times 200^2 = 2{,}127 \text{ pounds.}$$

The multiplication above can be carried out by the use of logarithms as follows:†

$$\begin{aligned} \log 0.00005427 &= \bar{5}.73456 \\ \log 392 &= 2.59329 \\ \log 2.5 &= 0.39794 \\ 2 \times \log 200 &= 4.60206 \\ \hline \log S &= 3.32785 \end{aligned}$$

Hence $S = 2{,}127$ pounds.

Problem 6.—The cylinder of a steam engine is 16 inches in diameter, and the length of the piston stroke 20 inches. The mean effective pressure of the steam on the piston is 110 pounds per square inch, and the number of revolutions per minute of the engine fly-wheel is 80. What is the power of the engine in indicated horse-power?

The formula for the horse-power of engine has been given in Chapter I, page 6:

$$H.P. = \frac{PLAN}{33{,}000} \qquad (2)$$

in which P = mean effective pressure in pounds per square inch,

* See MACHINERY'S Reference Series No. 40, Fly-Wheels, page 19, first edition.
† See MACHINERY'S Reference Series No. 53, Use of Logarithms and Logarithmic Tables.

L = length of stroke in feet,
A = area of piston in square inches,
N = number of strokes of piston per minute.

In the given problem $P = 110$; L (in feet) $= \frac{20}{12} = 1\ 2/3$; A, the area of the piston in square inches $= 16^2 \times 0.7854 = 256 \times 0.7854 = 201.06$; and N, the number of strokes of piston per minute $= 2 \times$ revolutions of fly-wheel $= 2 \times 80 = 160$. Substituting these values in the formula, we have:

$$H.P. = \frac{110 \times 1\ 2/3 \times 201.06 \times 160}{33{,}000} = 178.72.$$

Problem 7.—It is required to determine the diameter of cylinder and length of stroke of a steam engine to deliver 150 horse-power. The mean steam pressure is 75 pounds; the number of strokes per minute is 120. The length of the stroke is to be 1.4 times the diameter of the cylinder.

First insert in the horse-power Formula (2) the known values:

$$150 = \frac{75 \times L \times A \times 120}{33{,}000} = \frac{3 \times L \times A}{11}.$$

The last expression is found by cancellation.

Assume now that the diameter of the cylinder in inches equals D. Then $L = \frac{1.4\,D}{12} = 0.117\,D$, according to the requirements in the problem; the divisor 12 is introduced to change the inches to feet, L being in feet in the horse-power formula. The area $A = D^2 \times 0.7854$. If we insert these values in the last expression in our formula, we have:

$$150 = \frac{3 \times 0.117\,D \times 0.7854\,D^2}{11} = \frac{0.2757\,D^3}{11}$$

$$0.2757\,D^3 = 150 \times 11 = 1{,}650$$

$$D^3 = \frac{1{,}650}{0.2757}\ ;\ D = \sqrt[3]{\frac{1{,}650}{0.2757}} = \sqrt[3]{5984.8} = 18.15$$

The diameter of the cylinder, thus, should be about 18¼ inches, and the length of the stroke $18.15 \times 1.4 = 25.41$, or, say, 25½ inches.

Problem 8.—Find the horse-power required for compressing 10 cubic feet of air per second from 1 to 12 atmospheres, including the work of expulsion from the cylinder. Frictional and other losses are disregarded.

The formula for the work, W, in foot-pounds, required for compression and expulsion of 1 cubic foot of air from p_1 to p_n atmospheres is:

$$W = 3.463\,p_1 \left[\left(\frac{p_n}{p_1}\right)^{0.29} - 1\right] \times 14.7 \times 144 \qquad (12)$$

In the given problem $p_1 = 1$; $p_n = 12$; and as we are to compress 10 cubic feet instead of one, we must multiply the whole expression by 10. Thus:

$$W = 3.463 \times 1 \times \left[\left(\frac{12}{1}\right)^{0.29} - 1\right] \times 14.7 \times 144 \times 10$$
$$= 3.463 \times (12^{0.29} - 1) \times 14.7 \times 144 \times 10.$$

The value of the expression $12^{0.29}$ can be found only by the use of logarithms.*

$$\log 12 = 1.07918.$$
$$\log 12^{0.29} = 1.07918 \times 0.29 = 0.31296.$$
$$12^{0.29} = 2.056, \text{ and } 12^{0.29} - 1 = 1.056.$$

Hence:

$$W = 3.463 \times 1.056 \times 14.7 \times 144 \times 10 = 77{,}410 \text{ foot-pounds.}$$

This last result may be found by ordinary multiplication, or, more quickly, by logarithms as follows:

$$\begin{aligned} \log\ 3.463 &= 0.53945 \\ \log\ 1.056 &= 0.02366 \\ \log\ 14.7 &= 1.16732 \\ \log\ 144 &= 2.15836 \\ \log\ 10 &= 1.00000 \\ \hline \log W &= 4.88879 \qquad W = 77{,}410. \end{aligned}$$

As a horse-power equals 550 foot-pounds per second, the horse-power required for compressing 10 cubic feet of air from 1 to 12 atmospheres equals:

$$H.P. = \frac{77{,}410}{550} = 151 \text{ horse-power.}$$

Problem 9.—It is required to lift a weight weighing 1 ton by means of a screw having a lead of ½ inch. A lever passing through the head of the screw, and extending 4 feet out from the center, is provided at its outer end with a handle. How great a force must be applied at this handle to lift the required weight, friction being disregarded?

Let the weight to be lifted, in pounds, be W; the force applied at the end of the lever, F; the lead of the screw, l; and the length of the lever, in inches, r. The distance passed through by force F times this force must equal the distance weight W is lifted times the weight, or, expressed as a formula:

$$F \times 2r \times 3.1416 = W \times l. \qquad (13)$$

This formula is based on the fact that during one revolution of the screw and handle, force F acts through a distance equal to the circumference of the circle described by the handle, while the weight W is lifted an amount equal to the lead of the screw. If we insert the given values in the formula above, we have:

* See MACHINERY'S Reference Series No. 53, Use of Logarithms and Logarithmic Tables.

$$F \times 2 \times 48 \times 3.1416 = 2{,}000 \times \tfrac{1}{2}$$
$$F \times 301.59 = 1{,}000$$
$$F = \frac{1{,}000}{301.59} = 3.3 \text{ pounds.}$$

It will be seen that by the given arrangement a force of 3.3 pounds would be sufficient to lift a ton. Friction, however, has not been considered in this problem, and as the frictional resistance in machines using screws for conveying power is considerable, the actual force required would be a great deal more than 3.3 pounds.

Assume that is required to find the power if friction is considered. In this case we must know the diameter of the screw and the form of the thread. We will assume that the thread is square, and that the diameter of the screw is 3 inches. The depth of a ½-inch lead square thread is ¼ inch. The pitch diameter of the screw is, therefore, 3 — ¼ = 2¾ inches.

The formula for finding the force required at the end of the handle is:

$$Q = W \frac{f + \tan a}{1 - f \tan a} \times \frac{R}{r}$$

in which Q = force at end of handle, in pounds,
W = weight to be lifted = 2,000 pounds,
f = coefficient of friction,
a = angle of helix of the thread at the pitch diameter,
R = pitch radius of screw in inches = 1⅜ inch,
r = length of handle in inches = 48.

$$\tan a = \frac{\text{lead}}{3.1416 \times \text{pitch diam.}} = \frac{0.5}{3.1416 \times 2.75} = 0.058.$$

The coefficient of friction, f, may be assumed to be 0.15. If we now insert the known values in the formula, we have:

$$Q = 2{,}000 \times \frac{0.15 + 0.058}{1 - 0.15 \times 0.058} \times \frac{1.375}{48} = 12.02 \text{ pounds,}$$

or nearly four times as much as when friction was not considered.

Problem 10.—Determine the length of the main bearing of a large horizontal steam engine. The diameter of the crank-shaft is 10 inches, and the weight of the shaft, fly-wheel, crank-pin and other moving parts that may be supported by the bearings is 60,000 pounds. Assume that two-thirds of this weight, or 40,000 pounds, comes on the main bearing. The engine runs at 80 revolutions per minute.

The length of the main bearing of an engine may be found by the formula:*

$$L = \frac{W}{PK}\left(N + \frac{K}{D}\right) \qquad (14)$$

* See MACHINERY'S Reference Series No. 11, Bearings, page 11, first edition.

in which L = length of bearing in inches,
W = load on bearing in pounds,
P = maximum safe unit pressure on bearing at a very slow speed,
K = constant depending upon the method of oiling and care which the journal is likely to get,
N = number of revolutions per minute,
D = diameter of bearing in inches.

The safe unit pressure P for shaft bearings is 400 pounds; the factor K varies from 700 to 2,000. In this case, assume first-class care and drop-feed lubrication, in which case $K = 1{,}000$. The other quantities given are $W = 40{,}000$, $N = 80$, and $D = 10$.

Inserting these values in Formula (14), gives us:

$$L = \frac{40{,}000}{400 \times 1000}\left(80 + \frac{1000}{10}\right) = \frac{1}{10}(80 + 100) = 18 \text{ inches.}$$

Problem 11.—What is the carrying capacity of a helical spring having an outside diameter of 5 inches, made from ½-inch round steel? The tensile stress per square inch of section of spring must not exceed 80,000 pounds.

The formula for the carrying capacity of helical springs is:*

$$P = \frac{S\,d^3}{2.55\,D} \qquad (15)$$

in which P = safe carrying capacity,
S = safe tensile stress per square inch,
d = diameter of wire,
D = mean diameter of spring = outside diameter minus diameter of wire.

In the given problem $S = 80{,}000$; $d = ½$; and $D = 5 - ½ = 4½$. If these values are inserted in Formula (15) we have:

$$P = \frac{80{,}000 \times 0.5^3}{2.55 \times 4.5} = \frac{10{,}000}{11.475} = 871 \text{ pounds.}$$

Problem 12.—Find the weight of steam that will flow in one minute through a pipe 100 feet in length and 2 inches in diameter, if the initial pressure is 40 pounds (absolute) per square inch and the terminal or delivery pressure 35 pounds (absolute).

The formula for finding the weight of steam under the above conditions is:†

$$W = c\sqrt{\frac{w\,(P - P_1)\,d^5}{L}} \qquad (16)$$

in which W = pounds of steam per minute,
c = constant = 52.7 for a 2-inch pipe,

* See MACHINERY'S Data Sheet No. 22, July, 1903, Formulas for Coil Springs.
† See MACHINERY'S Data Sheet No. 109, March, 1909, Steam Pipe Sizes for Heating Systems.

w = weight per cubic foot of steam at initial pressure, in pounds,
P = initial pressure in pounds per square inch,
P_1 = terminal pressure in pounds per square inch,
d = diameter of pipe in inches,
L = length of pipe in feet.

In the present problem, $c = 52.7$; $w = 0.0972$ (obtained from tables in standard hand books); $P = 40$; $P_1 = 35$; $d = 2$; and $L = 100$. Inserting these values in Formula (16) gives:

$$W = 52.7\sqrt{\frac{0.0972 \times (40 - 35) \times 2^5}{100}} = 52.7\sqrt{0.1555} = 20.76 \text{ pounds.}$$

Problem 13.—Find the tractive power of a simple locomotive having 22-inch cylinder diameters, 26-inch stroke, a boiler pressure of 200 pounds, and 60-inch diameter driving wheels.

The formula for the tractive force of a locomotive is:*

$$T = \frac{0.85\,P\,d^2\,s}{D} \qquad (17)$$

in which T = tractive force in pounds,
P = boiler pressure in pounds per square inch,
d = diameter of cylinders in inches,
s = length of stroke in inches,
D = diameter of driving wheels.

Inserting the known values in Formula (17), gives:

$$T = \frac{0.85 \times 200 \times 22^2 \times 26}{60} = 35{,}655 \text{ pounds.}$$

Problem 14.—Find the diameter of the cylinders of a simple locomotive, having a tractive force of 30,000 pounds; length of stroke, 22 inches; diameter of driving wheels, 57 inches; and boiler pressure, 180 pounds.

The formula for the cylinder diameter is:†

$$d = \sqrt{\frac{T \times D}{P \times 0.85 \times s}} \qquad (18)$$

in which the letters denote the same quantities as in Formula (17).

If we insert the known values $T = 30{,}000$; $D = 57$; $P = 180$; and $s = 22$, in Formula (18), we have:

$$d = \sqrt{\frac{30{,}000 \times 57}{180 \times 0.85 \times 22}} = \sqrt{508.02} = 22.54 \text{ inches.}$$

or, approximately, 22½ inches diameter,

Problem 15.—Find the thickness of a cast iron cylinder to withstand a pressure of 1,000 pounds per square inch; the inside diameter of the cylinder is to be 10 inches, and the maximum allowable fiber stress per square inch 4,000 pounds.

* See MACHINERY'S Data Sheet No. 79, Constants for Calculating Tractive Force.
† See MACHINERY'S Reference Series No. 27, Locomotive Design, page 7

The thickness is found by the following formula:*

$$t=\frac{D}{2}\left(\sqrt{\frac{S+P}{S-P}}-1\right) \qquad (19)$$

in which t = thickness of cylinder wall in inches,
D = inside diameter of cylinder in inches,
P = working pressure in pounds per square inch,
S = allowable fiber stress in pounds per square inch

Inserting the given values in Formula (19), we have:

$$t=\frac{10}{2}\left(\sqrt{\frac{4000+1000}{4000-1000}}-1\right)=5\left(\sqrt{1.667}-1\right)=$$
$$5\ (1.29-1)=5\times 0.29=1.45, \text{ or say } 1\tfrac{1}{2} \text{ inch.}$$

Problem 16.—How many cubic feet of air does a disk fan, 30 inches in diameter, deliver when running at a speed of 500 revolutions per minute?

The answer to this problem is found by the following formula:†

$$C=0.6\,D\,R\,A, \qquad (20)$$

in which C = cubic feet of air delivered per minute,
D = diameter of fan in feet,
R = revolutions per minute,
A = area of fan in square feet.

In the given problem D, in feet $=\frac{30}{12}=2.5$; $R=500$; and $A=D^2\times 0.7854=2.5^2\times 0.7854=4.909$. Inserting these values in Formula (20), we have:

$$C=0.6\times 2.5\times 500\times 4.909=3{,}681.75 \text{ cubic feet.}$$

Problem 17.—What should be the weight of an 8-foot mean diameter fly-wheel, in pounds, for a two-cylinder, single-acting gas engine of 120 brake horse-power used in an electric lighting plant with continuous current generators, if the engine makes 300 revolutions per minute?

The following formula,‡ by Mr. R. E. Mathot, may be used for solving this problem:

$$P=K\frac{10.75\,N}{D^2\,a\,n^3} \qquad (21)$$

in which P = the weight of the rim, without arms or hub, in tons,
K = coefficient varying with the type of engine = 21,000 for a two-cylinder single-acting engine,
N = brake horse-power of engine,
D = mean diameter of fly-wheel, in feet,
a = amount of allowable variation = 1/50 for electric lighting by continuous current,
n = number of revolutions per minute.

* See MACHINERY'S Reference Series No. 17, Strength of Cylinders, page 21, first edition.
† See MACHINERY'S Reference Series No. 30, Fans, Ventilation and Heating, page 24.
‡ See MACHINERY'S Reference Series No. 40, Fly-Wheels, page 20, first edition.

In the given problem, where $K = 21{,}000$, $N = 120$, $D = 8$, $a = 0.02$, and $n = 300$, we have:

$$P = 21{,}000 \times \frac{10.75 \times 120}{8^2 \times 0.02 \times 300^2} = 0.78 \text{ ton.}$$

Expressed in pounds the weight of the rim equals $0.78 \times 2{,}000 =$ 1,560 pounds.

Problem 18.—Find the thickness of the piston for a steam engine having a cylinder diameter of 20 inches and a length of stroke of 24 inches.

The following formula may be used for finding the thickness of the piston:*

$$T = \sqrt[4]{L \times D} \qquad (22)$$

in which T = thickness of piston in inches,
L = length of stroke in inches,
D = diameter of cylinder in inches.

Inserting the given values in this formula, we have:

$$T = \sqrt[4]{24 \times 20} = \sqrt[4]{480}.$$

The fourth root of 480 can be most easily found by logarithms.†

$$\log T = \frac{\log 480}{4}.$$

$\log 480 = 2.68124$; $2.68124 \div 4 = 0.67031$.
$\log T = 0.67031$; $T = 4.68$ inches.

Problem 19.—Find the average horse-power required for taking a chip in a lathe 5/16 inch deep with a feed of 5/32 inch per revolution. The material cut is a bar of 30-point carbon steel, 4 inches in diameter, and is turned at a speed of 40 revolutions per minute.

A formula for finding the horse-power for turning in a lathe, based upon the experiments of Hartig, is as follows:‡

$$H.P. = 0.035 \times 3.1416 \times D \times n \times d \times t \times 0.28 \times 60 \qquad (23)$$

in which $H.P.$ = horse-power required for turning,
D = mean diameter of piece turned,
n = revolutions per minute,
d = depth of cut,
t = thickness of chip = feed per revolution.

In the problem given, D = outside diameter minus depth of cut = $4 - 5/16 = 3\ 11/16$; $n = 40$; $d = 5/16$; and $t = 5/32$. If we insert these values in the given formula, we have:

$H.\ P. = 0.035 \times 3.1416 \times 3.6875 \times 40 \times 0.3125 \times 0.1562 \times 0.28 \times 60 =$ 13.3.

Problem 20.—What horse-power may safely be transmitted by a 3 inches wide, machine-cut spur gear of 16-inch pitch diameter having 64 teeth, made of cast iron and running at a velocity of 120 revolutions per minute?

* See MACHINERY'S Data Sheet No. 120, Steam Engine Design.
† See MACHINERY'S Reference Series No. 53, Use of Logarithms and Logarithmic Tables.
‡ See MACHINERY'S Reference Series No. 16, Machine Tool Drives, page 29, first edition.

The formulas for the solution of this problem are as follows:*

$$V = 0.262\,D\,R \tag{24}$$

$$S = S_s \times \frac{600}{600 + V} \tag{25}$$

$$W = \frac{S\,F\,Y}{P} \tag{26}$$

$$H.P. = \frac{W\,V}{33{,}000} \tag{27}$$

in which V = velocity in feet per minute at pitch diameter,
D = pitch diameter in inches,
R = revolutions per minute,
S = allowable unit stress of material at given velocity,
S_s = allowable static unit stress of material,
W = maximum safe tangential load, in pounds, at pitch diameter,
Y = factor dependent upon pitch and form of tooth,
F = width of face of gear,
P = diametral pitch.
$H.P.$ = horse-power transmitted,

The known values to be inserted in the given formulas are $D = 16$, $R = 120$, S_s (for cast iron, assumed) $= 6{,}000$, $F = 3$; Y (for 64 teeth, standard form) $= 0.36$; and $P = 64 \div 16 = 4$. If we insert these values, as required, in the Formulas (24) to (27), and insert the values obtained in each formula in the next succeeding one, we have:

$$V = 0.262 \times 16 \times 120 = 503 \text{ feet.}$$

$$S = 6{,}000 \times \frac{600}{600 + 503} = 3{,}264 \text{ pounds per square inch.}$$

$$W = \frac{3{,}264 \times 3 \times 0.36}{4} = 881 \text{ pounds.}$$

$$H.P. = \frac{881 \times 503}{33{,}000} = 13.4 \text{ horse-power.}$$

Problem 21.—The initial absolute pressure of the steam in a steam engine cylinder is 120 pounds; the length of the stroke is 26 inches, the clearance 1½ inch, and the period of admission, measured from the beginning of the stroke, 8 inches. Find the mean effective pressure.

The mean effective pressure is found by the formula:

$$p = \frac{P\,(1 + \text{hyp.}\log R)}{R} \tag{28}$$

in which p = mean effective pressure in pounds per square inch,
P = initial absolute pressure in pounds per square inch,

* See MACHINERY'S Reference Series No. 15, Spur Gearing, page 29, second edition.

R = ratio of expansion, which in turn is found from the formula:

$$R = \frac{L + C}{l + C} \qquad (29)$$

in which L = length of stroke in inches,
l = period of admission in inches,
C = clearance in inches.

The given values are $P = 120$; $L = 26$; $l = 8$; and $C = 1\frac{1}{2}$. By inserting the latter three values in Formula (29), we have:

$$R = \frac{26 + 1\frac{1}{2}}{8 + 1\frac{1}{2}} = \frac{27.5}{9.5} = 2.89.$$

If we now insert the value of P and the found value of R in Formula (28), we have:

$$p = \frac{120\ (1 + \text{hyp. log } 2.89)}{2.89}.$$

The hyperbolic logarithm (hyp. log.) must be found from tables giving its value.* The hyperbolic logarithm for 2.89 is 1.0613. Inserting this value in our formula, we have:

$$p = \frac{120\ (1 + 1.0613)}{2.89} = \frac{120 \times 2.0613}{2.89} = 85.6 \text{ lbs. per square inch.}$$

Problem 22.—It is required to pump 12 cubic feet of water per minute with a centrifugal pump, raising it 35 feet, 15 feet by suction and 20 by discharge pressure. What will be the diameter of suction and discharge pipe required?

According to a formula by Fink:

$$d = 0.36 \sqrt{\frac{Q}{\sqrt{2g\ (h + h_1)}}} \qquad (30)$$

in which Q = quantity of water, in cubic feet, pumped per minute,
g = acceleration due to gravity = 32.16,
h = height of suction in feet,
h_1 = height of discharge in feet,
d = diameter of suction and discharge pipe, in feet.

Inserting the known values in Formula (30) we have:

$$d = 0.36 \sqrt{\frac{12}{\sqrt{2 \times 32.16\ (15 + 20)}}} = 0.36 \sqrt{\frac{12}{47.45}} = 0.36 \times 0.5 = 0.18,$$

approximately.

A pipe 0.18 foot, or $2\frac{1}{4}$ inches, in diameter would be required.

Problem 23.—What is the average pressure on the tool when turning hard cast iron, taking a chip $\frac{1}{8}$ inch deep with 1/16 inch feed per revolution?

* See MACHINERY'S Reference Series No. 53, Use of Logarithms and Logarithmic Tables.

The formula given by F. W. Taylor for finding the pressure on the tool is:*

$$P = CD^{\frac{14}{15}} F^{\frac{3}{4}} \qquad (31)$$

in which P = average pressure on tool in pounds,

C = a constant = 69,000 for hard cast iron,

D = depth of cut in inches,

F = feed per revolution, in inches.

Inserting the known values in this formula, we have:

$$P = 69{,}000 \times 0.125^{\frac{14}{15}} \times 0.062^{\frac{3}{4}}$$

To find the values of the two last expressions in the product above, we must make use of logarithms.† The whole product is also most easily found by means of logarithms.

$$\log 0.125 = \bar{1}.09691; \quad \frac{14}{15} \times \bar{1}.09691 = \bar{1}.15712$$

$$\log 0.062 = \bar{2}.79239; \quad \frac{3}{4} \times \bar{2}.79239 = \bar{1}.09429$$

$$\log 69{,}000 = \underline{4.83885}$$

$$\log P = 3.09026$$

Hence, P = 1,231 pounds.

Problem 24.—Find the diameter of shaft required to transmit 60 horse-power at 300 revolutions per minute, if the maximum safe stress of the material of which the shaft is made is 10,000 pounds per square inch.

The formula for finding the diameter of shaft is:

$$d = \sqrt[3]{\frac{321{,}400 \times H.P.}{RS}} \qquad (32)$$

in which d = diameter of shaft in inches,

$H.P.$ = horse-power to be transmitted,

R = revolutions per minute,

S = safe shearing stress of material of which shaft is made.

If we insert the given values in Formula (32), we have:

$$d = \sqrt[3]{\frac{321{,}400 \times 60}{300 \times 10{,}000}} = \sqrt[3]{6.428} = 1.86 \text{ inch.}$$

The diameter of the shaft may, therefore, be made, say, 1⅞ inch diameter.

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* See MACHINERY, June, 1907, engineering edition, page 568.

† See MACHINERY'S Reference Series No. 53, Use of Logarithms and Logarithmic Tables.

CHAPTER III

PRINCIPLE OF MOMENTS AS APPLIED TO THE LEVER*

The lever is the simplest element of a machine, and the principles of its action are of a simple nature. There is no reason why anyone who chooses to devote a little time to study should not be able to master these principles, and having done this, he will have gone a long way toward mastering the principles of all the elements that make up a machine.

Webster defines a lever as "a bar of metal, wood or other substance, used to exert a pressure or to sustain a weight, at one point at its length, by receiving a force or power at a second, and turning at a third on a fixed point called a fulcrum. It is of three kinds, according as either the fulcrum *F*, the weight *W*, or the power *P*, respectively, is situated between the other two." This is the usual definition of a lever as it is found in most books on mechanics and physics, and attention should be called to certain points about it that could easily lead a beginner astray and cause confusion at the outset. It is always best to start with a clear idea of a subject, so that there will be no uncertainty to begin with.

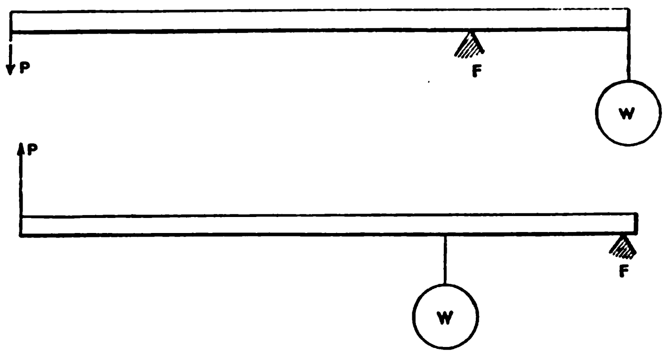
In Fig. 1 is a lever, in which, according to the definition, *W* is a weight acting at one point, *P* is the power or force acting at another point to raise the weight *W*, as indicated by the arrow, and *F* is the fulcrum on which the lever turns. That part of the lever between the weight and the fulcrum is called the "weight arm," and that part between the fulcrum and the power is called the "power arm." It will be noted that the fulcrum in Fig. 1 is located between the weight and power. In Figs. 2 and 3, however, are two levers in which the arrangement is different, the weight being placed between the power and fulcrum in Fig. 2, and the power placed between the weight and fulcrum in Fig. 3. These three figures illustrate the first, second, and third kinds of lever, as above defined.

The objections to this definition of the lever are, in the first place, the use of the word "power" for the force applied at the end of the lever to raise the weight. "Power" has a totally different meaning from "force," and takes into account not only force, but time and distance. A force is merely a push or pull, such as is exercised by the hand, and this is the kind of effort that is always required to raise a weight or overcome any other resistance. In the reference letters of the illustrations, therefore, we will let *P* stand for a push or a pull, as the case may be, instead of for the word "power." Hereafter, also, instead of calling the resistance to be overcome the "weight," we will

* MACHINERY, October and November, 1898.

call it the "resistance" and represent it by the letter *R*. A lever may have to overcome a number of resistances besides that of raising a weight, such as the resistance of friction, of a coiled spring, or of the pressure of steam, and the term "resistance" implies this better than the term "weight."

Finally, regarding the three kinds of levers mentioned above, there is no necessity for trying to separate levers into any number of classes, or for trying to remember to which class they belong in the solution of examples. All levers depend upon the same principles, which are simple



Figs. 1 and 2

and easily understood, and all that is necessary is to first master these principles without regard to the relative position of the applied force, the resistance, or the fulcrum.

The Moment of a Force

We have seen what is meant by the term "force," and the next thing to learn is what the moment of a force is. When a force acts at a point on a lever, that is, when that point is given a push or a pull, the tendency is to cause the lever to turn about its fulcrum. This tendency

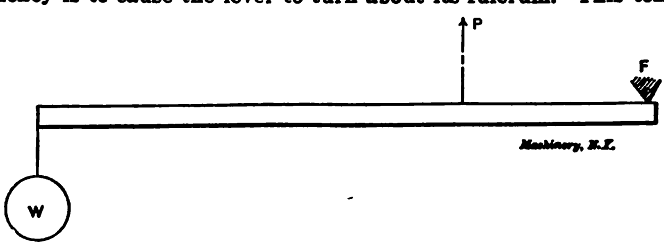


Fig. 3

depends first upon the strength of the force acting and second upon the perpendicular distance from the line of action of the force to the fulcrum. If either the strength of the push or pull exerted by the force, or the perpendicular distance of its line of action from the fulcrum, is changed, the tendency of the force to rotate the lever will be greater or less, as the case may be. The rotative effect of any force thus depends upon both the strength and the distance, and is measured

by their product, this product being called the moment of the force. The moment of force, therefore, is the measure of the turning effect of that force, and is found by multiplying the force by the perpendicular distance from its line of action to the fulcrum. If the force be measured in pounds and the distance in feet, the moment will be in foot-pounds; if the force be in pounds and the distance in inches, the moment will be inch-pounds; if the force be in tons and the distance in feet, the moment will be in foot-tons, etc. The foot-pounds measurement is the most commonly used, however.

This subject of moments is important—in fact, the most important in the whole subject of levers—and in order to fix it firmly in the mind, it will be helpful to have some common fact or operation that will illustrate it, and that can be referred to in solving complicated examples in which the application of the principle may not be entirely clear.

There is one kind of lever that is very familiar to every mechanic, and that is the wrench. We will select the wrench, therefore, to illustrate the subject of moments, and having once grasped the principle as applied to the wrench, no mechanic will be likely to have trouble with its other applications.

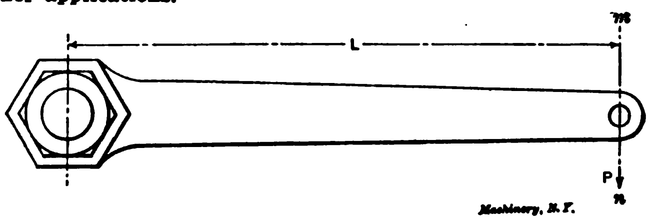


Fig. 4

Fig. 4 represents a box wrench, and, as is often done in work of a heavy character, a hole is punched in the outer end of the handle, into which a chain or rope can be hooked or fastened to assist in screwing the bolt or nut "home." Suppose the wrench is being used to screw up a nut, as shown in Fig. 4, and that the pull P on the rope is in the direction shown by the arrow, or in the direction of the line $m\,n$. The tendency of this pull to turn the wrench and nut will then be measured by the pull P in pounds, multiplied by the distance L in feet measured from the fulcrum at the center of the bolt, to the line $m\,n$, the distance being taken in the direction of a line at right angles or perpendicular to the line $m\,n$. This product gives the effect of the pull P in foot-pounds, and is called the moment of this force. Thus, if the pull P is 300 pounds, and the length L is 4 feet, the moment of the force P is $300 \times 4 = 1{,}200$ foot-pounds, and this is the measure of the turning effect of this force.

The reason why this is so will be evident if we consider another case shown in Fig. 5. Here the wrench has been placed in a new position, ready for another turn, and the pull P acts in the same direction as before, along the line $m\,n$. Now, anybody who has used a wrench knows that with the same pull a greater effect will be pro-

duced with the wrench as placed in Fig. 4 than as placed in Fig. 5, although in each case the hook is at the same distance (4 feet) from the fulcrum *F*. The direct distance, however, of the point of application of the force from the fulcrum does not necessarily have any influence on the effectiveness of this force in moving the lever. The only distance that can be considered is the perpendicular distance from the line along which the force acts to the fulcrum, and this distance is greater in Fig. 4 than in Fig. 5, and in the former the force of 300 pounds has a greater leverage than in the latter. In Fig. 5 the measure of the rotative effect is the pull *P*, which is 300 pounds, times the distance *L*, which in this case measures 2 feet, or $300 \times 2 = 600$ foot-pounds. The distance *L*, as before, is measured at right angles to

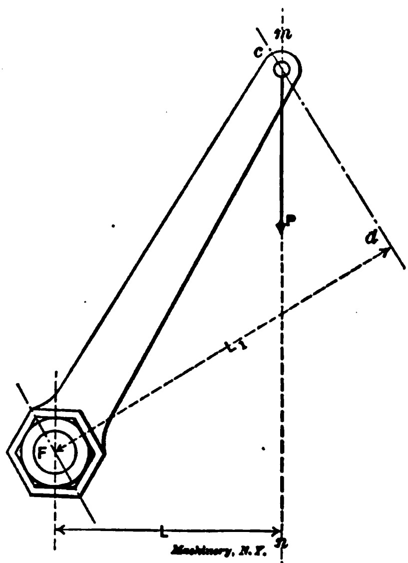


Fig. 5

the line *m n*, and if the rope had extended along the line *c d*, instead of the line *m n*, *L* would have been measured at right angles to the line *c d*, as indicated by the line L_1.

The True Lever Arm

The distance *L* in Figs. 4 and 5 is called the lever arm. Ordinarily the arm of a lever is understood to mean that part of the lever that lies between the fulcrum and the point where the force is applied, or between the fulcrum and the point where the resistance takes place; and such it is in a strict sense if the lever arm is straight and the force acts at right angles to the lever. But in Fig. 5 the true length of the lever arm is the distance *L*, and not the length of the handle of the wrench, because *L* is the effective length acting, in the position shown. *The true lever arm*, therefore, *is the perpendicular distance from the line of action of the force to the fulcrum.*

A familiar example of the moment of a force is to be had in the action of the foot in pedaling a bicycle. When the crank has passed the upper center, and the foot is ready for the downward push, it will require a much greater effort to drive the wheel ahead than when the crank is at right angles to the direction of the motion of the foot. The crank, of course, is of the same length whatever its position; but considered as a lever, the length of its arm varies from nothing at the upper center, to the full length of the crank at the extreme forward movement of the foot. The moment of the force exerted by the foot, therefore, gradually increases from nothing at the upper part of the stroke to the greatest amount at the forward position.

Still another illustration is to be had in the curved crank shown in Fig. 6. The crank turns about the point *F*, and a rod is attached at

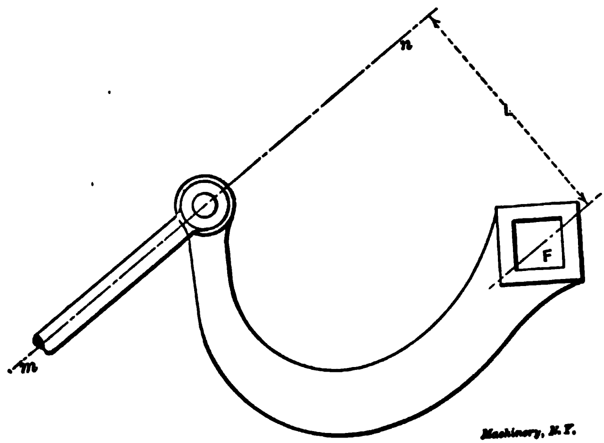


Fig. 6

the outer end which pushes in the direction shown by line *m n*. Drawing this dotted line *m n* through the point at which the push is applied and in the direction in which the push is exerted, we have *L*, which is drawn at right angles to *m n*, as the length of the lever arm, and the moment of the force is the length *L* multiplied by the force *P*.

The Principle of Moment

Thus far the illustrations that have been used have pertained to what might be called single-armed levers. We have considered only the forces acting without regard to the resistance that had to be overcome, and the levers themselves have been more of the nature of a crank than of a lever, though it is not always easy to make a distinction between the two. It is evident, however, that wherever a force is exerted, there must also be a resistance, as otherwise no initial force would be required to create motion. In the case of the wrench, the resistance was the friction between the threads of the bolt and nut acting at the end of a lever arm equal to the radius of the bolt; and

in the case of the bicycle crank, the resistance was at the rim of the bicycle wheel, the lever arm in this case being more complicated because of the sprockets and chain.

In Fig. 7 is shown a bell-crank lever pivoted at the fulcrum *F*. A pull *P* is exerted along the rod at the left, and this is balanced by another pull along the rod at the right, which acts as a resistance to the force *P*. To determine the relative rotative effects of the pull *P* and the resistance *R*, we must determine the moments of these two forces. To find the moment of *P*, draw a line *m n* through the point of the lever at which *P* takes effect, and in the direction of the line in which it acts. Then draw the line *L* from the fulcrum *F* and at right angles to the line *m n*. This will be the true lever arm, and the

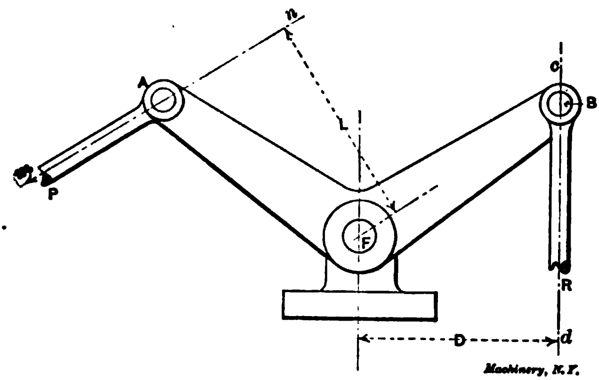


Fig. 7

moment of *P* will be the product of *P* and the length *L*. To find the moment of *R*, draw the line *c d* through the point of application of *R* and in the direction of *R*. Then draw the line *D* of a length equal to the perpendicular distance from *F* to line *c d*. This will be the true lever arm for *R*, and the moment of *R* will be the product of *R* and the distance *D*.

Since the moment of *P* measures the rotative effect of this force and the moment of *R* measures the rotative effect of the resistance, it is clear that if the lever is to balance, these two moments must be equal. If *L* is longer than *D*, as it is in this case, then *R* must be enough greater than *P* to make up for this, or otherwise the lever would begin to turn about *F*. This, in substance, is all there is to the principle of moments. The principle states that, if a body is to be in equilibrium, the sum of the moments of the forces which tend to turn it in one direction about a point is equal to the sum of the moments that tend to turn it in the opposite direction about the same point. In other words, if a body is to balance about a point, the opposing moments must be equal.

Calculation of Simple Levers

We will now be ready to solve examples of the lever by the aid of the principle of moments, and we will first consider that the weight

of the lever may be neglected, and that there are only two forces acting—the push or pull—which is applied to the lever, and the resistance overcome, these being balanced, of course, by the pressure at the fulcrum, which, in reality, is another force, but which need not be considered for the present, at least.

In Fig. 8 is shown a lever supported on the fulcrum F. At one end a push, P, of 10 pounds, is exerted, and at the other end is a resistance R, in the shape of a 100-pound weight. The distance from F to P is 40 inches, and from F to R, 4 inches. The principle of moments states that when a lever is in balance, the moment of the force tending to turn it in one direction must equal the moment of the force tending to turn it in the opposite direction. In Fig. 8 the moment of force P about fulcrum F, tending to depress the left-hand end of the lever, is $10 \times 40 = 400$ inch-pounds, and the moment of force R is $100 \times 4 = 400$ inch-pounds also, so that the lever is in balance.

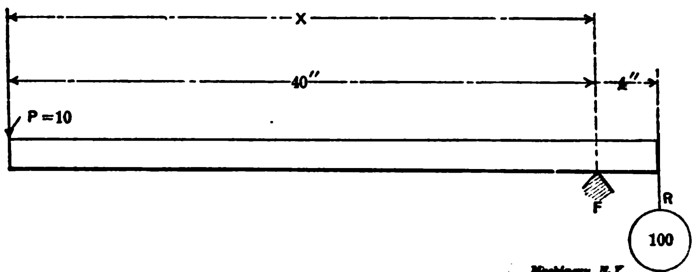


Fig. 8

Now, suppose that we had P, R, and the distance from F to R given in Fig. 8, and that we wanted to find the distance from F to P, which we will call x. By the principle of moments we have,

Moment of $P = 10 \times x$,

Moment of $R = 100 \times 4 = 400$.

But these moments are equal; hence, $10 \times x = 400$, and what we have to do is to find the value of x. It is clear that, if ten times the distance $x = 400$, the distance x must be 1/10 of 400, and all we have to do is to divide 400 by 10, giving 40 inches as the distance x.

Again, suppose it were desired to find the resistance R, the other quantities being known. For convenience we will take the moment of R first, because this contains an undetermined value. This is always a good rule to follow.

Moment of $R = 4 \times R$. (It makes no difference whether the 4 or the R is written first, but it is usual to write the figure first.)

Moment of $P = 10 \times 40 = 400$,

Then $4 \times R = 400$, and $R = \dfrac{400}{4} = 100$ pounds.

These simple examples contain all that need be known to solve lever problems where there are only two forces acting; but to make the subject still clearer, a more general example will be taken.

In Fig. 9 the lever shown is pivoted at F, which serves as the fulcrum. A push P is exerted by the rod at the right, which receives its motion from the cam and roller, as indicated. This push acts to overcome a resistance R, which acts along the rod seen at the left, and which may be supposed to consist of the resistance of the spring coiled around the rod, and of any piece of mechanism that this rod may have to operate. Let it be required to find how great a push, P, is necessary to overcome a resistance, R, of 250 pounds. The first thing is to find the length of the true lever arms, since without these the moments cannot be determined. To do this, first draw lines through the points on the lever at which the forces act, and in the direction in which they act. Thus, the force P acts at the point C, and the line $D\,H$ indicates the position and direction of this force. Likewise the force R

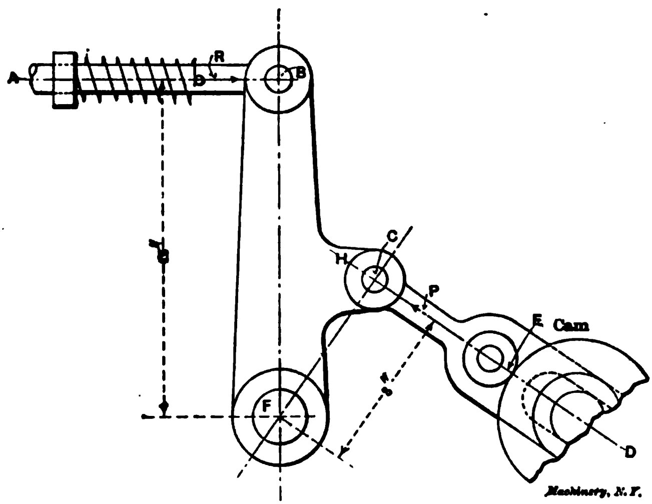


Fig. 9

acts at point B, and line $A\,B$ indicates the position and direction of force R.

Now, the lever arm of force P is the perpendicular distance from F to line $D\,H$, and the lever arm of force R is the perpendicular distance from F to line $A\,B$. Assume that these distances measure 8 and 16 inches, respectively. Then,

Moment of $P = 8 \times P$.

Moment of $R = 250 \times 16 = 4{,}000$.

$$8 \times P = 4{,}000; \text{ and } P = \frac{4{,}000}{8} = 500 \text{ pounds.}$$

Example.—Suppose $P = 400$, $R = 150$, and the short arm $= 6$ inches. What is the length of the long arm? Answer—16 inches.

The safety valve in Fig. 10 is an example of a lever in which there are three forces to be considered, if we take into account the weight

of the lever, which is quite essential to do. The valve at V is acted upon by the pressure of the steam, tending to raise it. This pressure constitutes the push P upon the lever, which is resisted by the suspended weight R, and the weight of the lever, which we will call R_1. The weight of the lever is effective at the point G, the center of gravity of the lever. This point can be found by balancing the lever on a knife edge, the center of gravity being directly over the knife edge. The fulcrum of the lever is at F, and the lever arms for R, R_1 and P are marked A, B, and C, respectively.

Example 1.—Assume that $A = 30$ inches, $B = 14$ inches, $C = 3$ inches,

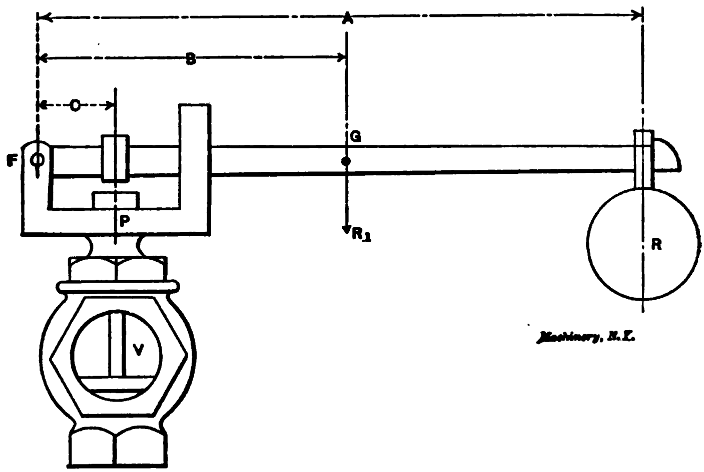


Fig. 10

$R = 20$ pounds, and $R_1 = 8$ pounds. Find what pressure of steam the valve will carry.

Moment of $P = 3 \times P$,
Moment of $R = 20 \times 30 = 600$,
Moment of $R_1 = 8 \times 14 = 112$.

For the valve to balance, the moment of P must be equal to the sum of the moments of R and R_1, for the moment of P tends to raise the lever, and the other moments tend to hold it down. Adding the moments of R and R_1, therefore, we have $600 + 112 = 712$, and this must balance the moment of P or $3 \times P$. Hence, $3 \times P = 712$, and $P = \frac{712}{3}$ $= 237\ 1/3$ pounds. This last part of the operation is like the work of the previous examples. The 237 1/3 pounds is the total pressure upon the valve, and to obtain the pressure per square inch that can be carried, we have simply to divide 237 1/3 by the area of the valve. To be theoretically exact, the weight of the valve and stem should be added to the figure 237 1/3.

Example 2.—Suppose it were desired to carry a total pressure upon the valve of 300 pounds. With the other dimensions remaining as

before, how heavy a weight R would have to be provided? Again, taking moments, we have,

$$\text{Moment of } R = 30 \times R,$$
$$\text{Moment of } R_1 = 8 \times 14 = 112,$$
$$\text{Moment of } P = 300 \times 3 = 900.$$

The sum of the first two moments must equal the last one, but we cannot add them as they stand, because we do not yet know what the first one is. Hence we will indicate the addition as follows:

$$30 \times R + 112 = 900.$$

Those who have had a little practice with formulas will have no trouble with finding the value of R; but for the benefit of those who have not, it can be said that we subtract the 112 from 900 and proceed as in the other examples. Thus, $900 - 112 = 788$, whence

$$R = \frac{788}{30} = 26\ 4/15 \text{ pounds.}$$

The following explanation will make the reason for subtracting 112 from 900 clear. We have found that the moment of R is 788; of R_1, 112; and of P, 900. Now, if 788 added to 112 equals 900, 900 must be 112 greater than 788, and 788 must be equal to 900 with 112 subtracted from it. Again, taking the formula as we have it, if $30 \times R$ plus 112 equals 900, $30 \times R$ must equal 900 with 112 subtracted from it.

Calculation of Compound Levers

It often happens that it is necessary to use two or more levers connected one to the other in a series, where it would not be convenient to obtain the desired multiplication with a single lever, or where it is necessary to distribute the forces acting. In such cases the levers are called compound levers, and their application is found in testing machines, car brakes, printing presses, and many other machines and devices. Probably the most familiar example is that of a pair of scales, and we will take this to illustrate the method of making the calculations for compound levers.

In Fig. 11 is a diagram showing an arrangement of levers that might be used for platform scales. The fulcrums of the various levers are in each case marked F. The scale platform is at E, bearing at each end on levers C and D, and loaded at the center with 1,000 pounds. A pressure of 500 pounds, therefore, is transmitted to lever C at a point 6 inches from the fulcrum, and 500 to lever D. As lever D is proportioned exactly the same as that part of lever C to the left of the center line of the weight—that is, as the distance from F to L in each case is exactly 4 feet, and the short arms are each 6 inches long—it follows that the final effect is the same as though the whole 1,000 pounds acted at a point 6 inches from the fulcrum F of the lever C.

Continuing through the various connections, the right-hand end of C pulls down on the lever B at a point 8 inches from its fulcrum, and this in turn pulls down on the scale beam A at a point 4 inches to the left of its fulcrum, and lifts the weight R. Question: What weight at R is required to balance the 1,000 pounds on the platform, assuming

that the system of levers is in balance so that there is no unbalanced weight to be considered? This is always provided for by a counterpoise on the scale beam.

The best way to solve any example of compound levers is to first determine the number of multiplications of each lever. Lever *A* has arms 40 and 4 inches long, and multiplies 10 times; lever *B* multiplies 4 times; and lever *C*, 25 times. Each lever multiplies in the same direction; that is, it tends to increase the force acting when we start at point *R*. Hence, the total multiplication is $10 \times 4 \times 25 = 1{,}000$, and

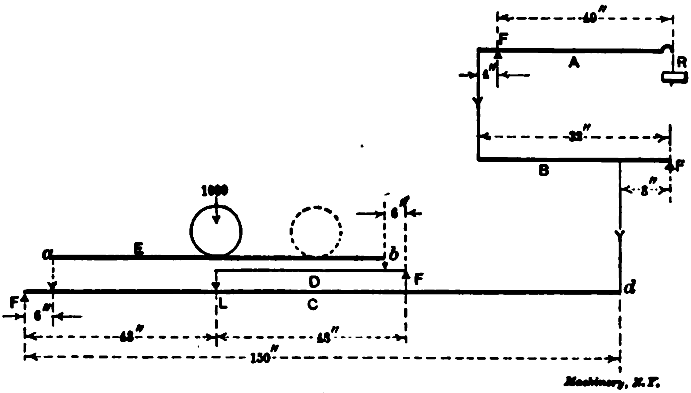


Fig. 11

thus one pound at *R* would balance the 1,000 pounds on the platform.

It may be asked whether with this arrangement the weighing of the scale would not be altered should the weight be moved to the dotted position shown in Fig. 11. A little thought will show that it would not. We have seen that the reduction from both points *a* and *b* to point *d* is 25 to 1, and it can make no difference whether 500 pounds acts at both *a* and *b*, or whether, for example, 300 pounds acts at *a* and 700 at *b*, the total 1,000 pounds being reduced 25 to 1 in either case.

CHAPTER IV

THE CENTER OF GRAVITY*

The force of gravity is exerted upon every one of the particles composing a body. The number of gravity forces acting upon a body may therefore be considered equal to the number of particles composing it. The sum or resultant of these individual forces constitutes the aggregate gravity of the body; and that point in the body at which may be applied a single resultant force that will have an effect the same as that of all the gravity forces acting upon its separate particles, is the center of gravity of the body. The center of gravity of a body will, therefore, be given by the position of the resultant of all the gravity forces acting upon its particles. If a body is supported upon its center of gravity, it will be in equilibrium in any position, and will have no tendency to rotate. This is, in substance, a definition that is sometimes given for the center of gravity.

Each one of the gravity forces acting upon the particles of a body, except those forces whose lines of action pass through its center of gravity, is producing a moment, and has a rotative effect. The lever arm of each moment is the perpendicular distance between the line of action of the force and the center of gravity of the body. Every such moment tends to produce rotation in the body, and as rotation is not produced when the body is supported upon its center of gravity, it follows that the center of gravity of a body is that point at which the moments of all the gravity forces acting upon its particles balance each other, or, in other words, at which the resultant moment of all the gravity forces is zero. This fact may be made use of in determining the position of the center of gravity. Different methods are employed for finding the center of gravity, according to the form of the body, or the arrangement of the system of bodies, for which it is to be found. Some of these methods will now be explained.

Center of Gravity of Lines

The word line, as here used, means a material line; that is, a homogeneous body of given length, having a uniform and very small transverse section, such as a fine wire. A theoretical line would, of course, have no width or thickness, and consequently, no mass and no gravity.

Single, Straight Line

The center of gravity of a straight line is at its middle point. If we conceive the line to be composed of uniform individual particles, the gravity of each particle will be the same; and the distance of each particle on one side of the middle point, from that point, will be the same as that of the corresponding particle on the opposite side.

* MACHINERY, September and October, 1898.

Hence, the moments of all the gravity forces acting upon the particles, taken about the middle point of the line, will balance, and that point will, therefore, be the center of gravity of the line. A straight line will balance upon its middle point; if supported upon that point, it will be in equilibrium in any position, and will have no tendency to rotate.

Two Straight Lines of Different Length

Let $A\ B$ and $C\ D$, Fig. 12, be two straight lines of any lengths and having any positions with respect to each other. The center of gravity of each line is at its middle point, as O and O_1. If these two centers of gravity be connected by the straight line $O\ O_1$ the center at gravity of the system will be somewhere on this line. Draw the line $O\ B_1$ equal and parallel to $O_1\ B = \frac{1}{2}\ A\ B$; on the opposite side of $O\ O_1$ lay off on the line $B\ A$, a length of $O_1\ C_1$ equal to $O\ C = \frac{1}{2}\ C\ D$, and draw B_1C_1. The point g, where the lines $O\ O_1$ and B_1C_1 intersect, will be

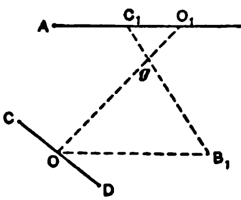
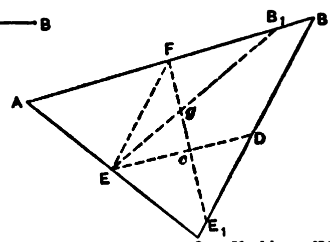


Fig. 12



Machinery, N.Y.

Fig. 13

the center of gravity of lines $A\ B$ and $C\ D$. If the given lines are parallel, $O\ B_1$ is simply laid off on $O\ D$ prolonged. The distance $O_1\ g$ may also be calculated; it is given by the equation:

$$O_1\, g = \frac{C\,D \times O\,O_1}{A\,B + C\,D}$$

Perimeter of the Triangle

Let $A\ B\ C$, Fig. 13, be any plane triangle, in which D, E and F are the centers of gravity of the three respective sides. Join any two of these centers, as D and E, and on this line determine, by the method just explained, the center of gravity c of the two sides joined. To do this, join E and F; the line $E\ F$ will be equal and parallel to $C\ D$; then lay off $D\ E_1$ equal to $C\ E$; the intersection c of the lines $D\ E$ and $E_1\ F$ will be the center of gravity of the sides $B\ C$ and $C\ A$. Now lay off $F\,B_1 = \frac{1}{2}\,A\,E + \frac{1}{2}\,B\,D$ and draw $E\,B_1$; the intersection g of the lines $E\ B_1$ and $c\ F$ will be the center of gravity of the three sides, or perimeter, of the triangle.

Circular Arc

Let $A\ B\ C$, Fig. 14, be the arc of a circle whose center is at O; $A\ C$ is the chord and B is the middle point of the arc. The center of gravity of the arc will be at some point g on the radius $O\,B$, at such distance from O that

$$O\,g = \frac{A\,C \times B\,O}{A\,B\,C}$$

Center of Gravity of Plane Surfaces

A theoretical surface has no thickness, and, therefore, no mass and no gravity. In mechanical problems, however, it is often necessary to find the center of gravity of a plane figure, or, more correctly, that point in its surface corresponding to what would be the center of gravity of the figure, were it a material body of uniform thickness. As here used, therefore, the word surface may be taken to mean a material surface, such as a very thin, homogeneous plate of a piece of cardboard.

Axis of Symmetry

If a plane figure can be divided by a straight line in such a manner that the two parts of the figure will exactly coincide when folded together along the line, the line so dividing the figure is called an

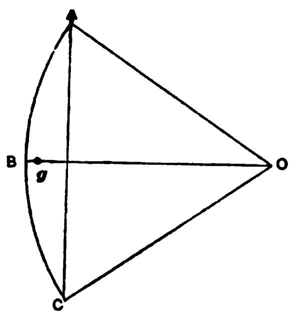


Fig. 14

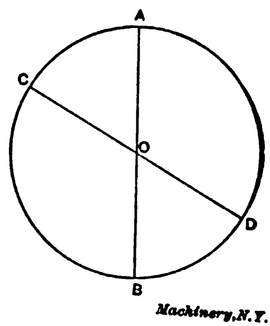


Fig. 15

axis of symmetry. The diameter of a circle and the diagonal of a square are axes of symmetry for those figures.

The center of gravity of a plane figure having an axis of symmetry, must lie on such axis; if the figure has more than one axis of symmetry, the center of gravity must be at the intersection of the axes. Let $A\,B$, Fig. 15, be a diameter of a circle whose center is at O; it is also an axis of symmetry, for, if folded along this diameter, the two parts of the circle will exactly coincide. If, now, we consider the area of the circle to be composed of straight lines perpendicular to $A\,B$, which are not shown in the figure, the diameter $A\,B$ will bisect each line; in other words, it will pass through the center of gravity of each line composing the area of the circle. Hence, the center of gravity of the entire system of lines composing the area of the circle, which will be the center of gravity of the circle itself, must be some point on the diameter $A\,B$. In like manner it can be shown that the center of gravity of the circle must lie on any other diameter, as the diameter $C\,D$. Consequently, the center of gravity of the circle must be at the center

O, the only point common to all diameters. That the center of gravity of the circle is at the geometrical center of the figure is so evident as to scarcely require proof; but the circle serves as a very simple example to illustrate the process of reasoning, which applies to any plane figure having two axes of symmetry, such as a circle, ellipse, rectangle, rhombus, equilateral triangle, square, or any regular polygon, and also to the perimeters of such figures.

Center of Gravity of Parts of Circles

Semicircle. The center of gravity is located on its axis of symmetry, at a distance of $0.4244r$ from the center of the circle, r being the radius of the circle.

Sector of a Circle. The center of gravity is located on its axis of symmetry, at a distance x from the center of the circle, the value of x being given by the equation:

$$x = \frac{2cr}{3l},$$

in which c is the chord and r the radius of the circle, and l the length of the arc.

Quadrant of a Circle. The center of gravity is located on its axis of symmetry, at a distance of $0.4244r$ from each radial side, or $0.6002r$ from the center of the circle, r being the radius of the circle.

Segment of a Circle. The center of gravity is located on its axis of symmetry, at a distance x from the center of the circle, the value of x being given by the equation:

$$x = \frac{c^3}{12a},$$

in which c is the chord and a the area of the segment.

Other Surfaces with Curved Outlines

Parabolic Surface. The center of gravity is located on its axis of symmetry, at 2/5 the length of the axis from the base.

Semi-parabolic Surface. The center of gravity is located at 2/5 of the length of the axis of the parabola from the base, and ⅜ the length of the semi-base from the axis.

Surface of a Hemisphere. The center of gravity is located at the middle of its axis or center radius.

Gravity Axis

It is not necessary, however, for a plane figure to have two, or even one, axis of symmetry, in order that its center of gravity may be determined. Any plane figure can be balanced upon a knife edge. The position of the knife edge will be defined by a straight line in such a position that the moments of all the gravity forces acting upon the particles composing the surface on one side of the line will just balance the moments of those on the other side. This line, about which the moments of the gravity forces balance, will here be called a gravity axis. By a process of reasoning analogous to that employed in finding

the center of gravity of the circle, it can be shown that every gravity axis of a plane figure contains the center of gravity of the figure. Consequently, the intersection of any two gravity axes determines the position of its center of gravity. It should be noticed that in many practical problems it is necessary to find the position of a gravity axis only, the exact center of gravity not being required.

Triangle

Let *A B C*, Fig. 16, be any triangle; the line *C D* extends from the vertex *C* to the middle of the opposite side. If we imagine the area of the triangle to be composed of straight lines parallel to the base *A B*, each of these parallel lines will be bisected by the line *C D;* that is, the line *C D* will pass through the center of gravity of each of the parallel lines. Every line composing the area of the triangle, and, consequently, the triangle as a whole, will just balance upon the line

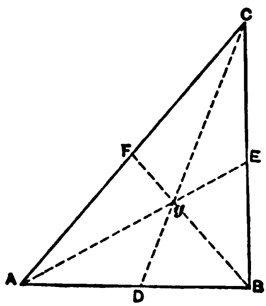


Fig. 16

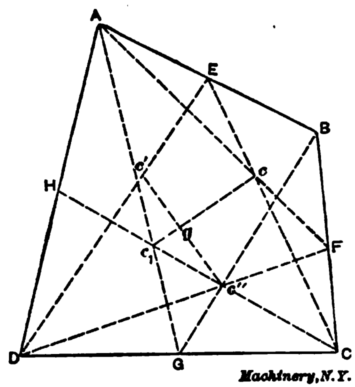


Fig. 17

C D, which will be a gravity axis of the triangle. If, also, a line be drawn from any other vertex of the triangle to the middle of the opposite side, as the line *A E*, it will also be a gravity axis. As the center of gravity must lie on both these gravity axes, it must be at their intersection *g*. It is not necessary, however, to draw more than one gravity axis, in order to determine the position of the center of gravity of a triangle. If a line be drawn from any vertex to the middle of the opposite side, the center of gravity of the triangle will be on this line and at two-thirds the length of the line from the vertex. Thus, the center of gravity *g*, Fig. 16, is at two-thirds the length of *A E* from *A*, two-thirds the length of *B F* from *B*, and two-thirds the length of *C D* from *C;* its position may be located on any one of the lines.

Trapezium

There are several quite satisfactory methods for finding the center of gravity of a trapezium. The following simple method is probably as expeditious as any, and, as it depends upon the method just explained for finding the center of gravity of a triangle, and is readily

connected with that method, it has the advantage of being easily remembered.

Let $A\ B\ C\ D$, Fig. 17, be any four-sided plane figure. Consider it first to be divided into the two triangles $A\ B\ C$ and $A\ D\ C$. The points E, F, G, and H are the centers of the respective sides, the common side $A\ C$ not being drawn. The intersection c of the lines $A\ F$ and $C\ E$ is the center of gravity of the triangle $A\ B\ C$, and, similarly, the intersection c_1 of the lines $A\ G$ and $C\ H$ is the center of gravity of the triangle $A\ D\ C$. The line $c\,c_1$ connecting these two centers of gravity, will be a gravity axis of the entire figure. The trapezium is then considered to be divided into the triangles $B\,A\,D$ and $B\,C\,D$, and, by a similar construction, the position of the gravity axis $c'\,c''$ is determined. The intersection g of these two gravity axes will be the center of gravity of the trapezium.

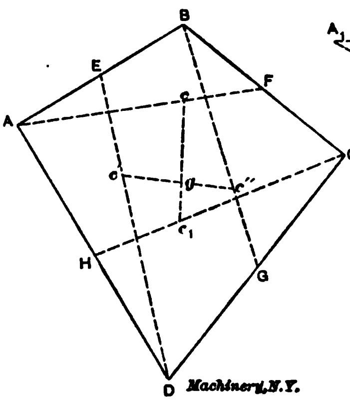


Fig. 18

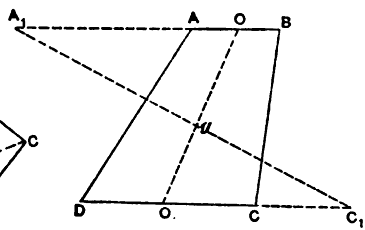


Fig. 19

For this construction, it is not necessary to draw the entire portion of each constructional line, as shown in the figure, but only such portions of the lines as are necessary to locate their intersections. Some may prefer the construction shown in Fig. 18; it is the same as that shown in Fig. 17, except that only one gravity axis is drawn for each triangle, and the center of gravity of the triangle located at two-thirds the length of the axis from its vertex.

Trapezoid

If the figure is a trapezoid, the following construction, taken from "Trautwine's Engineer's Pocket Book," is a very simple method of finding its center of gravity. Let $A\ B\ C\ D$, Fig. 19, be any trapezoid for which the center of gravity is to be found. Prolong the two parallel sides in opposite directions, making each prolongation equal to the other side, and join the extremities of the prolongations by a straight line; also join the centers of the parallel sides. The intersections of these lines will be the center of gravity of the figure. Thus, in the figure, $A\,A_1$ is made equal to $D\,C$, and $C\,C_1$ equal to $A\,B$, and the

extremities of the prolongations joined by the line A_1C_1, while the line $O\,O_1$ joins the centers of the parallel sides; the intersection g of the lines A_1C_1 and $O\,O_1$ is the center of gravity of the trapezoid.

Irregular Figure

The center of gravity of any irregular figure bounded by straight lines may be found by dividing it into triangles, finding the center of gravity of each triangle, and then finding the center of gravity of the system of triangles, the area of each being considered to be concen-

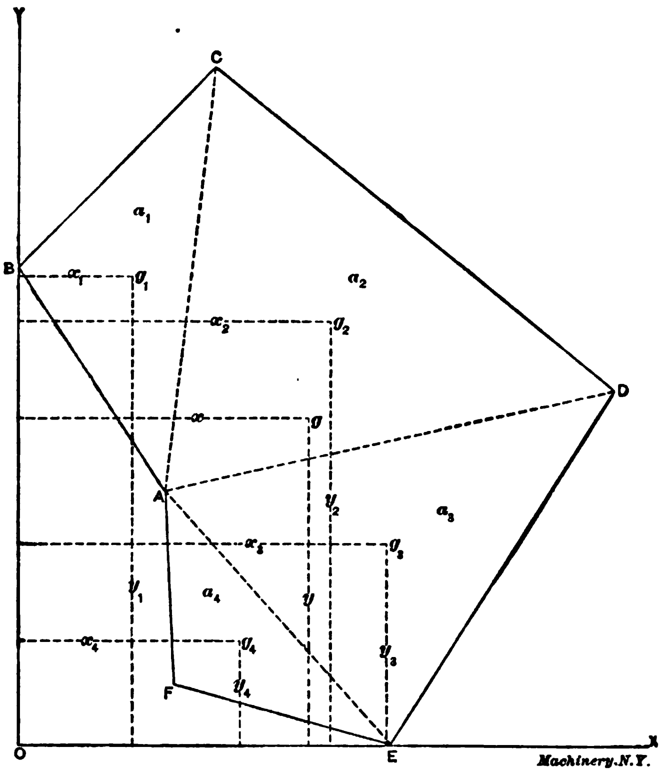


Fig. 20

trated at its center of gravity. For finding the center of gravity of the system of triangles, the method of rectangular co-ordinates may be employed. Let $A\,B\,C\,D\,E\,F$, Fig. 20, be any irregular figure bounded by straight lines. By the lines $A\,C$, $A\,D$, and $A\,E$ the figure can be divided into the four triangles $A\,B\,C$, $A\,C\,D$, $A\,D\,E$, and $A\,E\,F$, whose centers of gravity, g_1, g_2, g_3, and g_4 may be found by the method explained for triangles. Draw the vertical and horizontal axes $O\,Y$ and $O\,X$, intersecting at O; these may be any vertical and horizontal lines, but it is generally convenient to draw them through the left-hand and lower extremities of the figure, as shown; $O\,X$ is the axis of

abscissas and OY is the axis of ordinates. The lines x_1, x_2, x_3, and x_4 are, respectively, the abscissas of the centers of gravity from the axis of ordinates; and the line y_1, y_2, y_3, and y_4 are, respectively, the ordinates of the same points, or their perpendicular distances from the axis of abscissas. If a_1, a_2, a_3 and a_4 represent the areas of the four respective triangles, then the abscissa x to the center of gravity g of the entire figure will be given by the equation:

$$x = \frac{a_1x_1 + a_2x_2 + a_3x_3 + a_4x_4}{a_1 + a_2 + a_3 + a_4},$$

and the ordinate y to the same point will be given by the equation:

$$y = \frac{a_1y_1 + a_2y_2 + a_3y_3 + a_4y_4}{a_1 + a_2 + a_3 + a_4}.$$

This method applies to any figure or system of figures, either sepa rate or joined, that can be divided into triangles or other simpler figures such that their centers of gravity and areas can be determined. It will also apply to a system of weights or solid bodies.

Center of Gravity of Solid Bodies

The center of gravity of a sphere, spheroid, cylinder, cylindrical ring, cube, prism, parallelopipedon or any polyhedron is at the geometrical center of each body.

The center of gravity of a cylinder or prism is at the middle point of a line joining the centers of gravity of its parallel surfaces.

The center of gravity of a hemisphere is on its axis, or radius perpendicular to its base, at 3/8 length of the radius from the center of the sphere.

The center of gravity of a right cone or right pyramid is in the line joining the vertex with the center of gravity of the base, at 1/4 the length of the line from the base.

If a body be suspended freely at a point other than its center of gravity, its center of gravity will be vertically below the point of suspension. This principle affords an easy method of finding the center of gravity of any body, as described in the second method for finding experimentally the center of gravity of any plane figure.

CHAPTER V

THE FIRST PRINCIPLES OF THE STRENGTH OF BEAMS*

Having mastered the written engineering language sufficiently to deal successfully with formulas, the next step is to make the acquaintance of such engineering terms as are most frequently met with. Foremost among these are the terms relating to the strength of materials, and more especially the strength of beams.

If a bar is laid across two supports as in Fig. 21, and a weight placed in the center of it, we shall, if the bar be limber, witness the bending of the bar as shown, or as expressed in engineering terms, the deflection of the bar. It is obvious that the stiffer the bar, the less the deflection, and that a bar might be so lacking in stiffness as to actually break when the weight is placed upon it. Now the bar may lack stiffness from one or two causes; it may be that its dimensions are not well proportioned, or it may be made of soft and pliable materials. Sometimes both these causes are combined in the same bar. If the bar does not break when the weight is placed upon it, we must admit three facts; first, that the weight bends the bar; second, that the bar resists the bending; third, that the bar is able to resist the bending because it is large enough and made of stiff enough material.

Important Definitions

The bending effect that the weight has upon the bar is called the *bending moment* upon the bar due to the weight. The ability of the bar to resist the bending is called the *moment of resistance* of the bar. How these names first came into use the author does not know; perhaps there is no explanation, but the reader must not confuse the terms with any period of time because of the word moment. Time has nothing whatever to do with the strength of the bar, or the effect of the load upon it, except for such materials as wood, when loaded near to the limit of endurance.

In Fig. 21, the point at which the greatest bending occurs is directly under the weight, and we say the bending moment is maximum at this point, and the moment of resistance of the bar must equal the *maximum bending moment* at this point in the bar. In using the term bending moment, the engineer usually means the maximum bending moment, because this has the greatest bending effect upon the bar, and we shall hereafter drop the word maximum.

* MACHINERY, November, 1905.

Relation between Bending Moment and Moment of Resistance

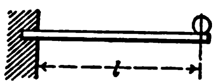
If now we let M = the bending moment on the bar, and R = the moment of resistance of the bar, we can express the relation of the two as given above thus:

$$M = R \tag{33}$$

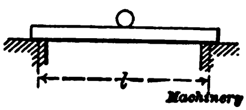
We said that the maximum bending moment was under the weight, and if the weight is placed further along on the bar, nearer one support than the other, the maximum bending moment will move with the weight. Also, if the bar is differently supported, the maximum bending moment will be at another point. For all cases of frequent occurrence, engineers have tables of formulas giving the position and amount of the maximum bending moment, so that it is only necessary to find in the tables the same case as the one we are considering, and

TABLE 2. BENDING MOMENT OF BEAMS UNDER VARIOUS SYSTEMS OF LOADING

W = total load.
l = length of beam in inches.
I = moment of inertia.
Z = section factor.

Case I 

Beam fixed at one end and loaded at the other.
Max. bending moment at point of support = $W\,l$.

Case II 

Beam supported at both ends. Single load in middle.
Max. bending moment at middle of beam $\dfrac{W\,l}{4}$.

taking the formula there given, substitute for the letters the corresponding dimensions in our case, and we have a numerical expression for the bending moment. The formulas given in these tables consist of combinations of dimensions measured along the bar, and weights of the loads on the bar. If, when substituting values for letters in the formula, loads are taken in tons, and distances in feet, the bending moment will be expressed in foot-tons, while if loads are taken in pounds and distances in inches, the usual custom, the bending moment will be expressed in inch-pounds.

Table 2 is a small portion of such a table as may be found in any book on machine design in any drafting-room or factory, as well as in all the handbooks issued by the steel mills.

So much for the first member of our equation, the bending moment on the bar. We have already seen that the bar offers resistance to bending by reason of two things: its dimensions, and the character of its material, and we should expect to find both dimensions and materials accounted for in the formula for the moment of resistance of any bar. This is just what the formula for the moment of resistance does. It is composed of two parts or terms, one of which expresses the resisting effect of the material of the bar, and the other

expressing the resisting effect possessed by the bar because of its shape and size. Let us investigate each term by itself, taking first the resisting effect of the material.

Tension and Compression Stresses

Let the reader take an ordinary rubber eraser of the form shown in Fig. 22, and bend it as shown in Fig. 23. While holding the eraser in the best position, draw a sharp knife across the top side. The cut immediately spreads out in the form of a V as shown at *a*. Draw the knife a second time through the same cut and the V spreads a little more. Now draw the knife across the bottom. The cut immediately closes up as at *b*. Draw the knife a second time across the same cut and it will still close up completely. In making the second cut on this side it may be necessary to release the eraser from the bent position, because the closing cut grips the knife blade and makes cutting difficult, but the cut will close, upon again bending the rubber.

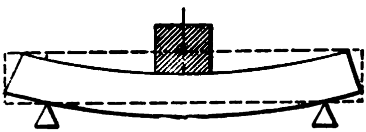


Fig. 21

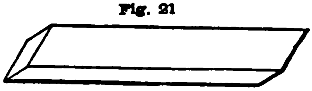


Fig. 22

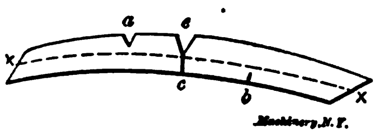


Fig. 23

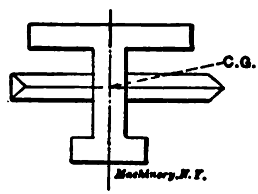


Fig. 24

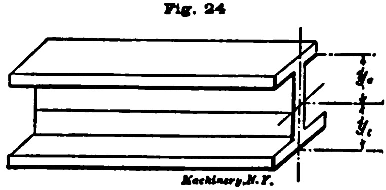


Fig. 25

Figs. 21 to 25

Having made the two cuts *a* and *b*, reverse the bend in the eraser and witness the closing of cut *a* and the opening of cut *b*. Now if you are a careful experimenter, you can start two such cuts as *a* and *b* directly opposite each other, and by cutting each one the same amount each time, you can succeed in bringing them nearly together in the center as shown at *c*. Of course, it will be impossible to bring them quite together, because that would cut the eraser apart, but by a little care you can satisfy yourself of these facts: that the portion of the eraser above the center line $x\,x$ separates when cut; and that the portion below the line closes when cut. Reversing the bend of the eraser as before reverses the behavior of the cuts, but observe that whichever way the eraser is bent, the opening cuts are to be found on the convex side, and the closing cuts on the concave side.

We know that all material (engineering and building material at least) is composed of fibers, and we must conclude from the behavior of our eraser that all the fibers on the convex side of the line $x\,x$

are stretched when the eraser is bent, while the fibers on the concave side of $x\,x$ are compressed. Since the cut through the stretched fibers opens like a V, we may conclude that those fibers lying at the top of the V are stretched more before the cutting than those lying at the point of the V. A careful examination of the cut made through the compressed fibers will show that at the outer portion of the cut, the edges are raised slightly, while at the inner portion, near the center of the eraser, the edges are not raised. We can account for this only by assuming that the fibers at the outer portion are more compressed than those near the center of the eraser.

Having performed these experiments and noted the results, we must admit the following facts: 1st, that half the fibers of a bent bar are in compression while the other half are stretched, or, as engineers say, are in tension; 2nd, that the amount of compression or tension is greatest at the outer portion of the bar, and diminishes towards the center of the bar; 3rd, that it follows from this, as well as from experiments with cut *c*, that there must be a line through the center of the bar where the fibers are neither in compression nor tension.

TABLE 3. STRENGTH OF MATERIALS—POUNDS PER SQUARE INCH.

MATERALS.	Ultimate Strength.	Safe Working Strength.	Factor of Safety.	Kind of Stress.
Cast Iron.	88,640	16,000	5	Compression
	15,620	8,000	5	Tension
Steel.	82,500	16.000	5	Compression
	80,000	16,000	5	Tension

Now the fibers resist any change in their condition, either stretching or compressing, the amount of resistance differing in different materials. Iron fibers more than rubber fibers for instance. When a bar is bent, engineers speak of the fibers as being under stress, some being under compressive stress, and others under tensile stress, as we have seen, and they speak of the bar as being subjected to fiber stress. Now, fiber stress is expressed in pounds per square inch, and it is the duty of the engineer when designing a beam or other structure to keep the fiber stress within safe limits, and these safe limits are given in hand books for a great variety of materials, in tables of which Table 3 is a sample.

Factors Determining the Moment of Resistance

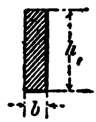
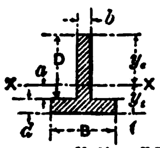
Since the material composing the bar derives its ability to resist bending by reason of the resistance of its fibers to changes, the fiber stress must be one of the terms expressing the moment of resistance of the bar. The fiber stress is denoted by the symbol f. The second term of the moment of resistance of the bar takes into consideration the strength the bar derives from its dimensions.

Bend the eraser in the direction of its greater thickness. We note it takes a much greater force to bend it thus than to bend it as we did at first, in the direction of its least thickness. If we repeat the

experiments with the cuts while bending the eraser thus, we shall find that everything witnessed before holds good for this case also. If we look for a reason for the greater force required to bend the eraser in the direction of its greater thickness, we shall find it in the fact previously observed, that the fibers are more stretched or compressed the further they are from the center line $x x$, and thus they present greater resistance to bending. The line $x x$ is called the *neutral axis*, because on it the fibers are neutral, being neither stretched nor compressed, and the fibers at the outer portion of the bar are called the extreme fibers, because they are furthest removed from the neutral axis $x x$.

The second term of the moment of resistance, taking account of the shape and size of the bar, is called the section factor, sometimes also

TABLE 4. VALUES OF I (MOMENT OF INERTIA) AND Z (SECTION FACTOR) FOR VARIOUS SECTIONS.

	Section.	I	Z	Area.
Case I.		$\frac{b h^3}{12}$	$\frac{b h^2}{6}$	$b h$
Case II.		$\frac{b y_2^3 + B y_1^3 - (B-b)a^3}{3}$	$\frac{I}{y_1}$ $\frac{I}{y_2}$	$Bd + bD$

Machinery, N.Y.

called the section modulus, Z, and is given in all hand books in the shape of tables for different shapes of beams, in the style shown in Table 4.

The neutral axis $x x$ is not always in the center of the bar, but it always passes through the center of gravity of the cross section of the bar.

Center of Gravity

Here we shall have to digress for a moment, since it is the intention to leave no term unexplained. The reader may best become acquainted with the center of gravity in the following manner: Cut out of stiff cardboard the shape of the cross section of the bar, and balance it over a sharp edge, in a manner as shown in Fig. 24. Draw a line across the card corresponding to the edge over which it is balanced. Repeat the experiment, turning the card around on the edge, and, balancing it a second time, draw another line. The intersection of these two lines will be the center of gravity of the section of the beam. If the experiment has been done with sufficient care, the card may be balanced upon a sharp point placed at the intersection of the two lines, just as if the entire material of the card were placed vertically above the point. A definition frequently met with is: The center of gravity is that point at which the entire weight of a body may be considered as concentrated.

Another way of finding the center of gravity is to suspend the card by a fine thread alongside of a plumb line, and when the card and line have come to rest, mark the position of the plumb line on the card. Turn the card around, and suspend a second time from a different point, and mark the position of the plumb line again. Where the two marks of the plumb line cross will be the center of gravity of the figure. No matter from how many points the card may be suspended, the plumb line will always be found to pass through the center of gravity. A line in the center of the beam directly opposite the center of gravity thus found will be the neutral axis.

Equation for Bending Moment

If we now take the equation expressing the relation of the bending moment on the bar to the moment of resistance of the bar, and use the symbols for the two parts of the moment of resistance, we shall have

$$M = R = f\,Z.$$

Some tables do not give the section factors Z for all sections directly, but say it is $\frac{I}{y}$, and therefore we must understand this expression.

The denominator y of the fraction is the distance from the neutral axis $x\,x$ to the extreme fiber of the bar, see Fig. 25, and the numerator I is what is called by engineers the *moment of inertia* of the section of the bar. Here again there is a chance for confusion because of the use of the word inertia.

Moment of Inertia

The term moment of inertia was originally employed when comparing the energies of rotating bodies. We know that a moving body possesses energy due to that property of matter which engineers call inertia. Inertia is not a force; it is simply resistance, and is due to the incapability of a dead body to move, or of a moving body to change its velocity or direction without the application of some external force. Now the number of foot-pounds of energy possessed by a moving body is equal to $\frac{1}{2} M V^2$, where M is the mass of the body, and V its velocity in feet per second. A moving body then, must be acted upon by an external force before it can be brought to rest. A rotating body is simply a very large number of particles moving in circular paths about an axis called the axis of rotation. Each moving particle, therefore, possesses energy due to its inertia, and the energy of each particle is equal to $\frac{1}{2}\,m\,v$, where m is the mass of the particle, and v its velocity in feet per second. But the energy varies as $m\,v^2$, because simply dividing by 2 does not change the relative values. It is also obvious that the circumferential velocity of each particle varies as the distance from the axis of rotation, which distance or radius we call r. Hence, substituting r for v, the energy of each particle varies as $m\,r^2$. Suppose we imagine that the whole mass of the rotating piece, that is, the sum of all the small particles m, is concentrated in a circle that is of such diameter that the energy pos-

sessed by the entire mass is the same as before. The radius of this imaginary circle is called the radius of gyration, and is usually designated by the letter r. Now we may say that $M r^2$, where M stands for the whole mass, is a measure of the energy of the rotating piece. This expression $M r^2$ is given the name moment of inertia, each particle of which the rotating body is composed possessing a turning moment about the axis of rotation, due to its motion and inertia.

When it was discovered that the flexure of a beam depended upon the value $a r^2$ (where a is the area of the cross section of the bar, and r^2 is the mean of the squares of the distances of the infinite number of small areas into which the area of the section may be supposed to be divided, from the center of gravity of the section) $a r^2$ was seen to be similar to the expression $M r^2$, which, in connection with the rotating bodies, had already become known as the moment of inertia; so, very carelessly on the part of those who first committed the error, it was said that the flexure of a beam varied as its moment of inertia, not because inertia has anything to do with it, for, of course, it has not, but because $a r^2$, the expression for the moment of resistance to flexure, happened to vary in the same way as the moment of inertia $M r^2$ of the same body when rotating about its center of gravity.

The moment of inertia of a bar may be calculated by several methods, but the table in hand books give it for all usual shapes of sections, and we will not attempt the calculation here.

Universal Formula for Bending Strength of Beams

Since we are sometimes able to find in tables only the moment of inertia of a bar, and not the section factor, we must bring our formula one step further, thus:

$$M = R = f Z = f \frac{I}{y} \tag{34}$$

or

$$Z = \frac{I}{y} = \frac{M}{f} \tag{35}$$

and here we have the formula for determining the size required for any beam.

For beams in which the center of gravity is not the center of the beam, there will be two values of y, one of which we will denote as y_c, being the distance from the neutral axis to the extreme fibers in compression, and the other as y_t, being the distance from the neutral axis to the extreme fibers in tension, see Fig. 25.

In some materials the ability of the fibers to resist tension is about equal to their ability to resist compression, while in other materials there may be great inequality in this direction, some being much stronger in tension than in compression, while others are stronger in compression than in tension. In such a material we shall have two values of f, which we will denote as f_c and f_t for compression and tension, respectively.

Ultimate and Safe Stresses

Some tables on the strength of materials give what is called the ultimate or breaking strength of the materials, while other tables give the safe working strength of materials.

When using the latter tables, the values given are to be substituted directly for f_c and f_t in the formula. Since it would not do to have the material of which a beam is made strained up to the breaking point, we must, when using the former tables, make use of a factor of safety. This factor of safety is a divisor by which the breaking strength of a beam is divided to allow a margin of strength in the beam. The divisor varies from 2 to 10, and the proper use of different divisors is given in the text books.

To illustrate, the breaking strength of steel may be given as 80,000 pounds per square inch, and $\frac{80{,}000}{5} = 16{,}000$. If we substitute 16,000 for f in the formula, we shall be working out our results with a factor of safety of 5, and the beam should not actually break until loaded with five times the load designed for. As a matter of fact, the beam would become badly bent long before five times the load could be placed upon it.

Limit of Elasticity

We have seen that all material deflects under the influence of a load, and we suppose that the elasticity of the material causes it to spring back to its original condition when the load is removed. This is true within limits, but there is a point somewhere between the safe load and the breaking load at which, when the load is gradually increased, the beam becomes strained beyond its power to return to its original condition upon the load being removed. This point is variously called the *limit of elasticity*, the *yield point*, the *point of permanent* set.

Practical Examples

Let us now take up two examples illustrating the ground we have just passed over, and the use of the tables.

Example 1. A rectangular steel bar, 2 inches thick, is built into a wall as in Fig. 26, and is to hold a load of 3,000 pounds at its outer end, 36 inches from the wall. We wish to know the required depth to make the beam.

1st. Consider the bending moment on the beam. According to Case 1, Table 2, the bending moment is

$$M = W\,l.$$

For our case we know W and we know l, and substituting these for the letters in the formula gives us

$$M = 36 \times 3{,}000 = 108{,}000 \text{ inch-pounds.}$$

2d. Consider the permissible fiber stress in the steel bar. Table 3 gives the safe working strength of steel as 16,000 pounds per square inch.

3d. Using Formula (35) we can find the value of the section factor for our beam. We know the bending moment and we know the fiber

stress, so substituting these for the letters in the formula we get

$$Z = \frac{M}{f} = \frac{108{,}000}{16{,}000} = 6.75.$$

4th. Find the section of our beam in Table 4, Case 1, where we find that the section factor is

$$Z = \frac{b\,h^2}{6}.$$

We know Z and we know b, so substituting these values for the letters, we get

$$6.75 = \frac{2 \times h^2}{6}.$$

If we multiply both sides of this equation by 6, we shall not change its value, but shall get

$$6 \times 6.75 = 2 \times h^2.$$

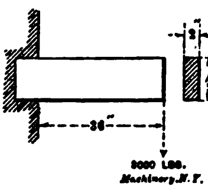


Fig. 26

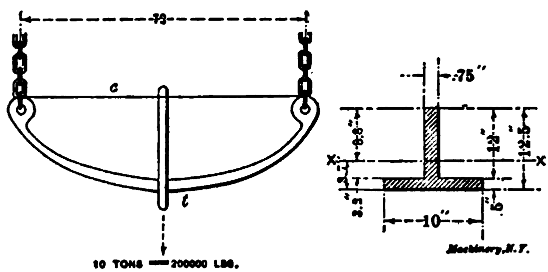


Fig. 27

If we now divide both sides by 2, we shall not change its value, but shall get

$$\frac{6 \times 6.75}{2} = h^2 = 20.25.$$

5th. We can most conveniently find the square root of 20.25 from a table of squares and roots which may be found in any hand book. This square root is 4.5, and we thus find that

$$h = 4.5 \text{ inches.}$$

If we make the beam 2 inches thick by 4.5 inches deep by 36 inches long, it will support a load of 3,000 pounds at its free end, and the fibers will be strained to 16,000 pounds per square inch.

Example 2. Let us undertake to design a suspension beam like Fig. 27 to carry ten tons, the material to be cast iron. The proposed section of the beam is more complicated than that of the previous example, and we cannot obtain a result quite so directly.

1st. Inspect the proposed beam to locate the compression and tension flanges. We find the compression flange is on top and the tension flange on the bottom, and we mark them c and t respectively.

2d. Table 3 shows us that cast iron is stronger in compression than in tension, hence we conclude that we should have more metal on the

tension side than on the compression side, and accordingly we place the section with the heavy side down.

3d. Assume a section by making the best guess possible as to the dimensions shown heavy in the figure. Cut out this section of cardboard, and find the location of the neutral axis $x x$ as previously explained. Now fill in the figures shown light by measuring the cardboard section.

4th. Find the section in Table 4. Here we find that before we can get the section factor of the beam we must get the moment of inertia of the beam. Substitute the dimensions of our section for the letters of the formula given in Table 4, and we shall get

$$I = \frac{(0.75 \times 8.8^3) + (10 \times 3.7^3) - (10 - 0.75)\ 3.2^3}{3}$$

$$= \frac{511.1 + 506.5 - (9.25 \times 3.2^3)}{3}$$

$$= \frac{1017.6 - 303.12}{3} = \frac{714.48}{3} = 238.$$

5th. Now divide the moment of inertia just found by the distances of the extreme fibers from the neutral axis, that is, by y_c and y_t, and we get

$Z_c = \dfrac{I}{y_c} = \dfrac{238}{8.8} = 27$, the section factor for the compression side.

$Z_t = \dfrac{I}{y_t} = \dfrac{238}{3.7} = 64.3$, the section factor for the tension side.

6th. Inspect Table 2 and find the bending moment on the beam according to Case 2; substituting the dimensions of the beam, and the load to be carried, in the formula given, we have

$$M = \frac{W\,l}{4} = \frac{20{,}000 \times 72}{4} = 360{,}000 \text{ inch-pounds.}$$

7th. Dividing the bending moment just found by the section factors found in the 5th step, will give the fiber stress on the beam according to Formula (35), thus

$\dfrac{360{,}000}{27} = 13{,}333$ pounds per square inch on the compression side.

$\dfrac{360{,}000}{64.3} = 5{,}600$ pounds per square inch on the tension side.

The latter is too high, so another guess must be made, making the section heavier on the tension side. Then the steps 3, 4, 5 and 7 must be repeated, and if the fiber stress then comes below 3,000 pounds per square inch, the section will be right.

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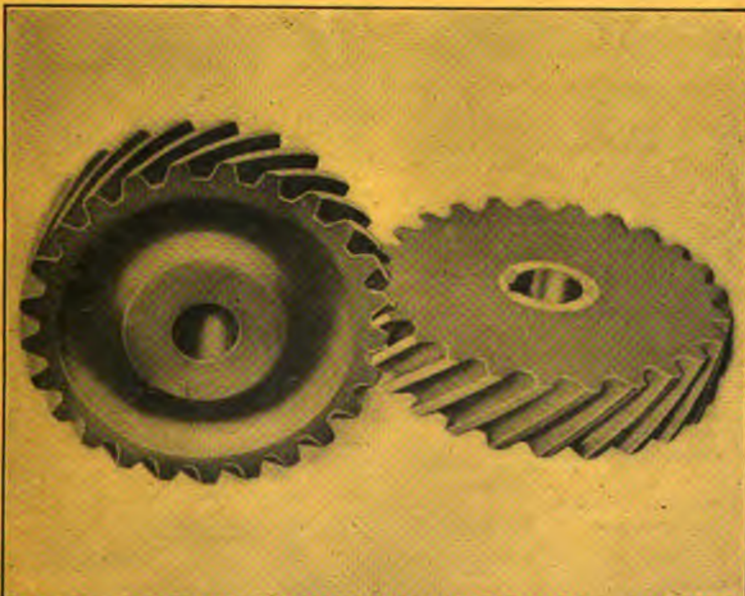

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NUMBER 20

SPIRAL GEARING

THIRD EDITION—REVISED AND ENLARGED

CONTENTS

49-55 Lafayette Street, New York City

The present—the third—edition of this number of MACHINERY'S Reference Series has been thoroughly revised, and a considerable amount of new matter has been included. Chapters on "Herringbone Gears" and on "Calculating Gears for Generating Spirals on Hobbing Machines" have been added, and the chapter on "Setting the Table when Milling Spiral Gears" has been entirely rewritten.

CHAPTER I

RULES AND FORMULAS FOR DESIGNING SPIRAL GEARS

In accordance with time-honored custom, this contribution to the art of designing helical or "spiral" gears opens with an apology. The subject is one which, from its very nature, can be approached by any one of a number of different ways, and it has been approached by so many of these possible different ways that perhaps the subject has become quite confused in the minds of many readers of technical literature. The writer does not offer the excuse of novelty in the methods presented in the following paragraphs, since some of the details, which were independently worked out by him have been described by others. His reason for adding one more to the series of solutions of helical gear problems is that the method described appears to reduce the most serious of this class of problems (Class II, page 9) to its simplest elements. The method of procedure will be described without proof or comment.

The terms "spiral gear" and "helical gear" are, in usage, synonymous, the only difference being that the former of these terms is absolutely incorrect. Inasmuch, however, as the word "spiral" is in such common use among mechanics in this connection, the writer has not had the moral courage to use the more proper term throughout this treatise, but it would be a good plan for the readers to become familiar with the term "helical" as applied to gearing.

Dimensions and Definitions

Some of the terms used will require explanation. The center angle of a pair of helical or spiral gears is the angle made by the two center lines or axes of the gears, as taken in a view perpendicular to both axes. In Fig. 1 are shown views of three sets of spiral gears taken in the plane which shows the center angle. At the left is the ordinary case in which the shafts are at right angles with each other, so that the center angle (γ) is 90 degrees. In the second case γ is less than 90 degrees, and in the example shown at the right it is more. It should be noted in the last two cases that the position of the shaft axes is identical, but that the two center angles are located on opposite sides of axis *A*. In order to know on which side of the center line to take the center angle in cases like those shown, we have to reckon with the position of the teeth of the gears in contact. The center angle is taken at the side which includes the line *x-x*, passing lengthwise of the teeth of the gears at the point of contact with each other. Since the teeth are laid out differently in the two cases, the angles are different. The case shown in the center is much the more usual of the two, the other being very rare.

In Fig. 2 is given a diagram showing what is meant by the "tooth angle" of a helical gear. In using the expression "tooth angle," the angle made by the tooth with the axis of the gear is meant, not the angle of the tooth with the face of the gear. Fig. 2 shows α_a as the tooth angle of gear *a*, and α_b as the tooth angle of gear *b*, used in the sense in which we will use them.

The number of teeth and the pitch diameter are terms which are identically the same as those used for spur gearing* and, therefore, require no explanation. Practically all spiral gears are of small size, and hence are reckoned on the diametral pitch rather than the circular pitch system. All the rules and formulas given will, therefore, make use of the diametral pitch only. This may easily be found from the circular pitch by dividing 3.1416 by the circular pitch. The center distance is, of course, the shortest distance between the axes, and so is measured along the perpendicular common to both of them.

The regular diametral pitch of a spiral gear will be found the same as for a spur gear by dividing the number of teeth by the pitch diam-

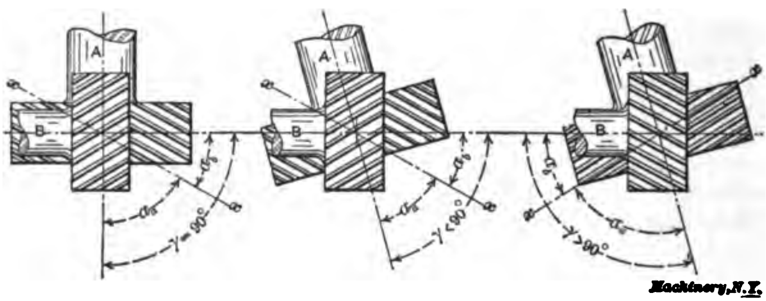


Fig. 1. Spiral Gears with Different Center Angles

eter in inches. We are not interested in knowing what this is, however, since it does not enter into the calculations at all and since the cutter used has to be for a somewhat finer diametral pitch. This is shown more clearly in Fig. 3. The normal diametral pitch, or diametral pitch of the cutter used, is reckoned from mesurements taken along the pitch cylinder at right angles to the length of the tooth. P' represents the regular circular pitch, while P_n' represents the normal circular pitch. The diametral pitch may be found from this by dividing 3.1416 by P_n'. This is the pitch of the cutter to be used. The cutter, as explained on page 5, cannot be selected for the actual number of teeth in the gear, but must take into account the helix angle of the teeth as well, since the curvature as measured on a line at right angles to the helix is at a greater radius than when measured on the circle.

The length of the helix, or the lead, as shown in Fig. 3, is the length of pitch cylinder required to permit one complete revolution of the tooth if the latter were carried around for the full length of this cylinder. In Fig. 4, the relation of lead, circumference, and tooth angle is plainly shown, the helix *AB* here being developed on a plane.

* See MACHINERY'S Reference Series No. 15, Spur Gearing, Chapter II.

The addendum S, and whole depth W of the tooth for helical gears is the same as for plain spur gears. The normal thickness of tooth at the pitch line, T_n, as shown in Fig. 3, is measured in a direction perpendicular to the face of the tooth. The regular tooth thickness is shown at T, but with this we are not concerned. The outside diameter, as for spur gears, is found by adding twice the addendum to the pitch diameter.

Rules for Calculating Spiral Gear Dimensions

The following rules are used for calculating the dimensions of spiral or helical gears:

Rule 1. *The sum of the tooth angles of a pair of mating helical gears is equal to the shaft angle;* that is to say, in Figs. 1 and 2, angle a_a added to a_b equals γ, as is self-evident from the engravings.

Rule 2. *To find the pitch diameter of a helical gear, divide the number of teeth by the product of the normal pitch and the cosine of the tooth angle.*

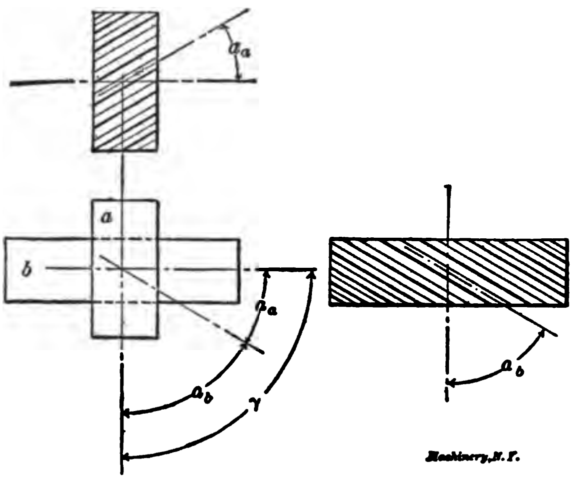


Fig. 2. Diagram Showing Notation used for Tooth Angles

Rule 3. *To find the center distance, add together the pitch diameters of the two gears and divide by 2.* This rule is evidently the same as for spur gears.

Rule 4. *To prove the calculations for pitch diameters and center distance, multiply the number of teeth in the first gear by the tangent of the tooth angle of that gear, and add the number of teeth in the second gear to the product; the sum should equal twice the product of the center distance multiplied by the normal diametral pitch, multiplied by the sine of the tooth angle of the first gear.*

Rule 5. *To find the number of teeth for which to select the cutter, divide the number of teeth in the gear by the cube of the cosine of the tooth angle.*

Rule 6. *To find the lead of the tooth helix, multiply the pitch diameter by* 3.1416 *times the cotangent of the tooth angle.*

The rules relating to the addendum and the whole depth of tooth are the same as for spur gears. They are:

Rule 7. *To find the addendum, divide* 1 *by the normal diametral pitch.*

Rule 8. *To find the whole depth of tooth space, divide* 2.157 *by the normal diametral pitch.*

Rule 9. *To find the normal tooth thickness at the pitch line, divide* 1.571 *by the normal diametral pitch.*

Rule 10. *To find the outside diameter, add twice the addendum to the pitch diameter.*

The problem of designing a pair of spiral gears presents itself in

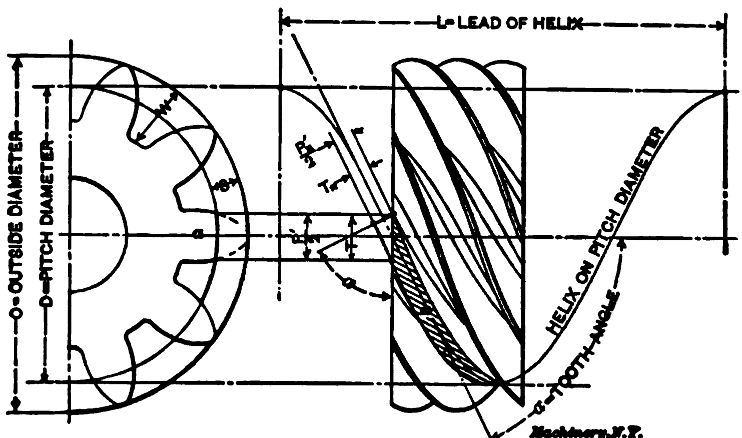


Fig. 3. Diagram of Spiral Gear, Illustrating Terms used in the Calculations

general in two different forms or classes, which may be stated as follows:

Class 1. The diametral pitch and the numbers of teeth in the two gears are given.

Class 2. A fixed center distance is given together with the velocity ratio or the numbers of teeth, with the requirement that standard cutters of even diametral pitch be used.

Examples of Calculations Under Class 1

Let it be required to make the necessary calculations for a pair of spiral gears in which the shafts are at right angles. Normal diametral pitch equals 3; number of teeth in gear equals 45; number of teeth in pinion equals 18.

There being no restriction in this particular case as to center distance we have to settle first on the tooth angles for the two gears. To obtain the highest efficiency, some authorities advise that the smallest tooth angle be given to the gear having the smallest number of teeth; and this angle should not, in general, run below 20 degrees. Keeping

it nearly 30 or even up to 45 would be better. On the basis $a_a = 30$, and $a_b = 60$ degrees, we have the following calculations:

To find the pitch diameters, use Rule 2:

$$\text{Pitch diameter of gear} = \frac{45}{3 \times \cos 60^\circ} = 30 \text{ inches.}$$

$$\text{Pitch diameter of pinion} = \frac{18}{3 \times \cos 30^\circ} = 6.928 \text{ inches.}$$

To find the center distance, use Rule 3:

$$\frac{30 + 6.928}{2} = 18.464 \text{ inches.}$$

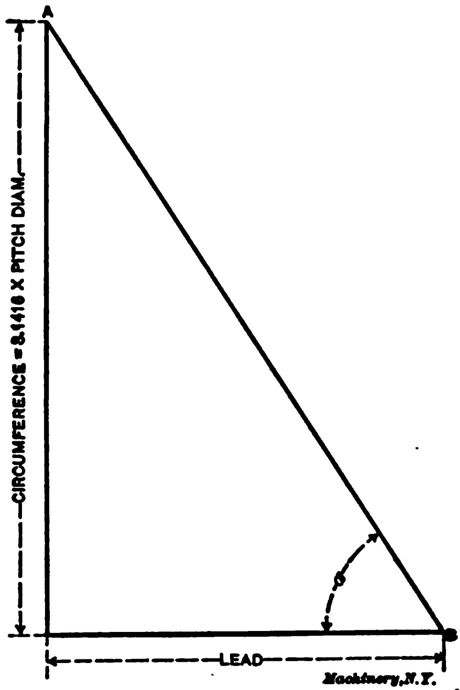


Fig. 4. Diagram Showing Relation between Pitch Diameter, Lead and Angle of Helix

To prove that the previous calculations are correct, use Rule 4:

$45 \times \tan 60^\circ + 18 = 95.940.$

$2 \times 18.464 \times 3 \times \sin 60^\circ = 95.939.$

These two results are so nearly alike that the previous calculations may be considered fully correct.

To find the number of teeth for which to select the cutter, use Rule 5:

$$\text{For gear, } \frac{45}{(\cos 60^\circ)^3} = 360.$$

For pinion, $\dfrac{18}{(\cos 30^\circ)^3} = 28$, approximately.

To find the lead of the tooth helix, use Rule 6:

Lead for gear $= 3.1416 \times 30 \times \cot 60^\circ = 54.38$ inches.

Lead for pinion $= 3.1416 \times 6.928 \times \cot 30^\circ = 37.70$ inches.

To find the addendum, use Rule 7:

$$\text{Addendum} = \frac{1}{3} = 0.333 \text{ inch.}$$

To find the whole depth of tooth space, use Rule 8:

$$\text{Whole depth} = \frac{2.157}{3} = 0.719 \text{ inch.}$$

To find the normal tooth thickness at the pitch line, use Rule 9:

$$\text{Tooth thickness} = \frac{1.571}{3} = 0.527 \text{ inch.}$$

To find the outside diameter, use Rule 10:

For gear, $30 + 0.666 = 30.666$ inches.

For pinion, $6.928 + 0.666 = 7.594$ inches.

This concludes the calculations for this example. If it is required that the pitch diameters of both gears be more nearly alike, the tooth angle of the gear must be decreased, and that of the pinion increased.

Suppose we have a case in which the requirements are the same as in Example 1, but it is required that both gears shall have the same tooth angle of 45 degrees. Under these conditions the addendum, whole depth of tooth and normal thickness at the pitch line would be the same, but the other dimensions would be altered as below:

$$\text{Pitch diameter of gear} = \frac{45}{3 \times \cos 45^\circ} = 21.216 \text{ inches.}$$

$$\text{Pitch diameter of pinion} = \frac{18}{3 \times \cos 45^\circ} = 8.487 \text{ inches.}$$

$$\text{Center distance} = \frac{21.216 + 8.487}{2} = 14.851 \text{ inches.}$$

Number of teeth for which to select cutter:

For gear, $\dfrac{45}{(\cos 45)^3} = 127$, approximately.

For pinion, $\dfrac{18}{(\cos 45)^3} = 51$, approximately.

Lead of helix for gear $= 3.1416 \times 21.216 \times \cot 45^\circ = 66.65$ inches.

Lead of helix for pinion $= 3.1416 \times 8.487 \times \cot 45^\circ = 26.66$ inches.

Outside diameter of gear $= 21.216 + 0.666 = 21.882$ inches.

Outside diameter of pinion $= 8.487 + 0.666 = 9.153$ inches.

Examples of Calculations Under Class 2*

In Class 2 the writer is going to make use of the term "equivalent diameter." The quotient obtained by dividing the number of teeth in a helical gear by the diametral pitch of the cutter used gives us a very useful factor for figuring out the dimensions of helical gears, so the writer has ventured to give it the name "equivalent diameter," an abbreviation of the words "diameter of equivalent spur gear," which more accurately describe it. This quantity cannot be measured on the finished gear with a rule, being only an imaginary unit of measurement.

Rule 11. *To find the equivalent diameter of a helical gear, divide the number of teeth of the gear by the diametral pitch of the cutter by which it is cut.*

The process of locating a railway line over a mountain range is divided into two parts; the preliminary survey or period of exploration, and the final determination of the grade line. The problem of designing a pair of helical gears resembles this engineering problem in having many possible solutions, from which it is the business of the designer to select the most feasible. For the exploration or preliminary survey, the diagram shown in Fig. 5 will be found a great convenience. The materials required are a ruler with a good straight edge, and a piece of accurately ruled, or, preferably, engraved, cross-section paper. If a point, *O*, be so located on the paper that *B O*, the distance to one margin line, be equal to the equivalent diameter of gear *a*, while *B′ O*, the distance to the other margin line, be equal to the equivalent diameter of gear *b*, then (when the rule is laid diagonally across the paper in any position that cuts the margin lines and passes through point *O*) *D O* will be the pitch diameter of gear *a*, *D′ O* the pitch diameter of gear *b*, angle *B O D* the tooth angle of gear *a* and angle *B′ O D′* the tooth angle of gear *b*. This simple diagram presents instantly to the eye all possible combinations for any given problem. It is, of course, understood that in the shape shown it can only be used for shafts making an angle of 90 degrees with each other.

The diagram as illustrated shows that a pair of helical gears having 12 and 21 teeth each, cut with a 5-pitch cutter, and having shafts at 90 degrees with each other and 5 inches apart, may have tooth angles of 36° 52′ and 53° 8′, and pitch diameters of 3 inches and 7 inches, respectively.

Suppose it were required to figure out the essential data for three sets of helical gears with shafts at right angles, as follows:

1st. Velocity ratio 2 to 1, center distance between shafts 2¼ inches.
2d. Velocity ratio 2 to 1, center distance between shafts 3⅜ inches.
3d. Velocity ratio 2 to 1, center distance between shafts 4 inches.

We will take the first of these to illustrate the method of procedure about to be described.

We have a center distance of 2¼ inches and a speed ratio between driver and driven shafts of 2 to 1. The first thing to determine is

* MACHINERY, May, 1906.

the pitch of the cutter we wish to use. The designer selects this according to his best judgment, taking into consideration the cutters on hand and the work the gearing will have to do. Suppose he decides that 12-pitch will be about right. In Fig. 5 it will be remembered that *D O* was the pitch diameter of gear *a*, while *D′ O* was the pitch diameter of gear *b*. That being the case, *D O D′* is equal to twice the distance between the shafts. In the problem under consideration this will be equal to 2 × 2¼, or 4½ inches. Fig. 6 is a skeleton outline showing the operation of making the preliminary survey with rule and cross-section paper. *A G* and *A G′* represent the margin lines of the sheet, while *D D′* represents the graduated straight-edge. By the

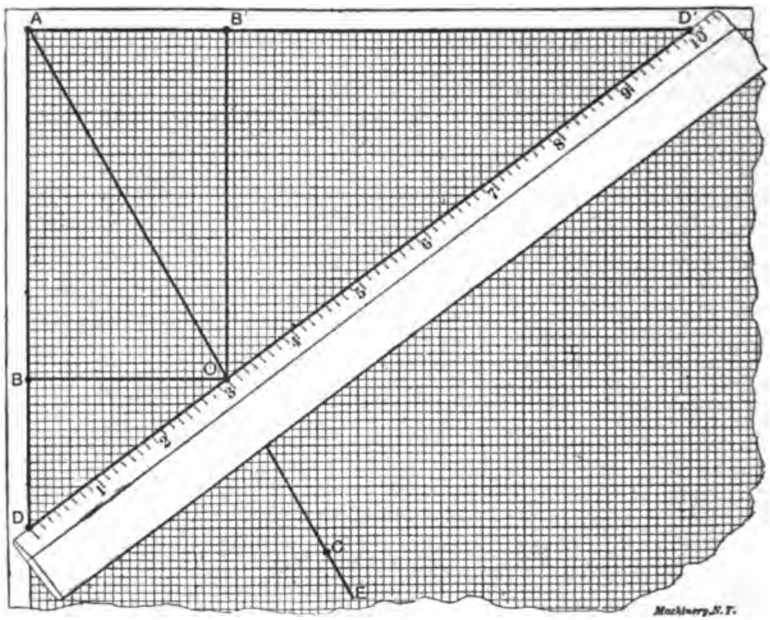


Fig. 6. Preliminary Solution with Rule and Cross-section Paper

conditions of the problem, the distance between points *D* and *D′*, where the ruler crosses the margin lines, must be equal to 4½ inches. There has next to be determined at what angle of inclination the ruler shall be placed in locating this line. To do this, we will first find our "ratio line." Select any point *C* such that *C F′* is to *C F* as 2 is to 1, which is the required ratio of our gears. Draw through point *C*, so located, the line *A E*. Line *A E* is then the ratio line, that is, a line so drawn that the measurements taken from any point on it to the margin lines will be to each other in the same ratio as the required ratio between the driving and driven gear. Now, by shifting the ruler on the margin lines, always being careful that they cut off the required distance of 4½ inches on the graduations, it is found that when the rule is laid as shown in position No. 1, cutting the ratio line at *O′*,

the distance from the point of intersection to corner A is at its maximum. For the minimum value, the tooth angle is the limiting feature. For a gear of this kind, 30 degrees is, perhaps, about as small as would be advisable, so when the ruler is inclined at an angle of about 30 degrees with margin line $A\ G'$, and occupies position No. 2 as shown, it will cut line $A\ E$ at O'', and the distance cut off from the point of intersection to corner A will be at its minimum value. The ruler must then be located at some intermediate position between No. 1 and No. 2.

Supposing, for example, 14 teeth in gear a and 28 teeth in gear b be tried. According to Rule 11, the equivalent diameter of gear a will then be $14 \div 12$, or 1.1666 inch; the equivalent diameter of b will be

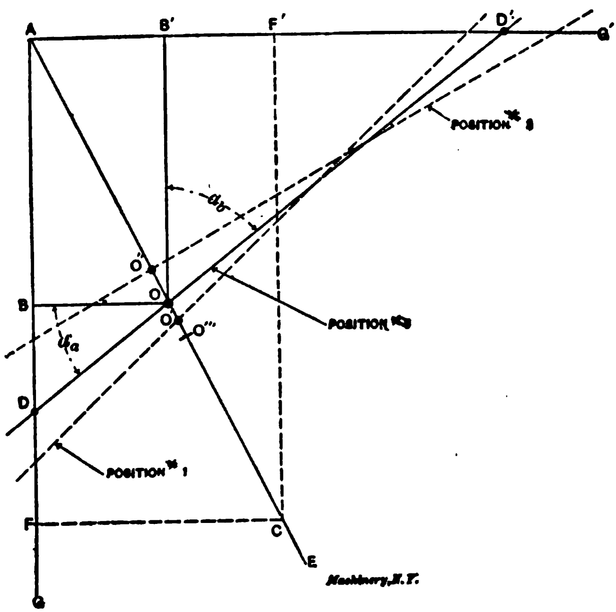


Fig. 6. Preliminary Graphical Solution for Problem No. 1

$28 \div 12$, or 2.3333 inches. Returning to the diagram to locate the point of intersection, it will be found that point O''' is so located that lines drawn from it to $A\ G$ and $A\ G'$ will be equal to 1.1666 inch and 2.3333 inches respectively, but this is beyond point O', which was found to be the outermost point possible to intersect with a 4½-inch line, $D\ D'$. This shows that the conditions are impossible of fulfillment.

Trying next 12 teeth and 24 teeth, respectively, for the two gears, the equivalent diameters by Rule 11 will be 1 inch and 2 inches. Point O is now so located that $O\ B$ equals 1 inch and $O\ B'$ equals 2 inches. Seeing that this falls as required between O' and O'', stick a pin in at this point to rest the straight-edge against, and shift the straight-edge about until it is located in such an angular position that the

margin lines AG and AG' cut off $4\frac{1}{2}$ inches, or twice the required distance between the shafts, on the graduations. This gives the preliminary solution to the problem. Measuring as carefully as possible, DO, the pitch diameter of gear *a*, is found to be about 1.265 inch diameter, and $D'O$, the pitch diameter of gear *b*, about 3.235 inches. Angle $B\ O\ D$, the tooth angle of gear *a*, measures about 37° 50′. Angle $B'\ O\ D'$, the tooth angle of gear *b*, would then be 52° 10′ according to Rule 1. To determine angle $B\,O\,D$ more accurately than is feasible by a graphical process, use the following rule:

Rule 12. *The tooth angle of gear a in a pair of mating helical gears, a and b, whose axes are 90° apart, must be so selected that the equivalent diameter of gear b plus the product of the tangent of the tooth angle of gear a by the equivalent diameter of gear a, will be equal to the product of twice the center distance by the sine of the tooth angle of gear a.* (This rule, it will be seen, is simply a modification of Rule 4.)

That is to say, in this case, $O\ B' + (O\ B \times$ the tangent of angle $B\,O\,D) = D\,D' \times$ the sine of angle $B\,O\,D$. Perform the operations indicated, using the dimensions which were derived from the diagram, to see whether the equality expressed in this equation holds true. Substituting the numerical values:

$$2 + (1 \times 0.77661) = 4.5 \times 0.61337,$$
$$2.77661 = 2.76016,$$

a result which is evidently inaccurate.

The solution of the problem now requires that other values for angle $B\ O\ D$, slightly greater or less than 37° 50′, be tried until one is found that will bring the desired equality. It will be found finally that if the value of 38° 20′ be used as the tooth angle of gear *a*, the angle is as nearly right as one could wish. Working out Rule 12 for this value:

$$2 + (1 \times 0.79070) = 4.5 \times 0.62024,$$
$$2.79070 = 2.79108.$$

This gives a difference of only 0.00038 between the two sides of the equation. The final value of the tooth angle of gear *a* is thus settled as being equal to 38° 20′. Applying Rule 1 to find the tooth angle of gear *b* we have: $90° - 38°\ 20' = 51°\ 40'$. The next rule relates to finding the pitch diameter of the gears.

Rule 13. *The pitch diameter of a helical gear equals the equivalent diameter divided by the cosine of the tooth angle;* (*or the equivalent diameter multiplied by the secant of the tooth angle*). This rule is a modification of Rule 2.

If a table of secants is at hand, it will be somewhat easier to use the second method suggested by the rule, since multiplying is usually easier than dividing. Using in this case, however, the table of cosines, and performing the operation indicated by Rule 13, we have for the pitch diameter of gear *a*:

$$1 \div 0.78442 = 1.2748, \text{ or } 1.275 \text{ inch, nearly;}$$

and for the pitch diameter of gear *b*:

$$2 \div 0.62024 = 3.2245, \text{ or } 3.225 \text{ inches, nearly.}$$

To check up the calculations thus far, the pitch diameter of the two gears thus found may be added together. The sum should equal twice the center distance, thus:

$$1.275 + 3.225 = 4.500,$$

which proves the calculations for the angle.

Applying Rule 10 to gear *a*:

$1.2748 + (2 \div 12) = 1.2748 + 0.1666 = 1.4414 = 1.441$ inch, nearly.

For gear *b*:

$3.2245 + (2 \div 12) = 3.2245 + 0.1666 = 3.3911 = 3.391$ inches, nearly.

In cutting spur gears of any given pitch, different shapes of cutters are used, depending upon the number of teeth in the gear to be cut. For instance, according to the Brown & Sharpe system for involute gears, eight different shapes are used for a gear from 12 teeth to a rack. The fact that a certain cutter is suited for cutting a 12-tooth spur gear is no sign that it is suitable for cutting a 12-tooth helical gear, since the fact that the teeth are cut on an angle alters their shape considerably. To find out the number of teeth for which the cutter should be selected, use Rule 5.

Applying Rule 5 to gear *a*:

$$12 \div 0.784^3 = 12 \div 0.4818 = 25 -,$$

and for gear *b*:

$$24 \div 0.620^3 = 24 \div 0.2383 = 100 +,$$

giving, according to the Brown & Sharpe catalogue, cutter No. 5 for gear *a* and cutter No. 2 for gear *b*.

In gearing up the head of the milling machine to cut these gears it is necessary to know the lead of the helix or "spiral" required to give the tooth the proper angle. To find this, use Rule 6. In solving problems by this rule, as for Rule 5, it will be sufficient to use trigonometrical functions to three significant places only, this being accurate enough for all practical purposes. Solving by Rule 6 to find the lead for which to set up the gearing in cutting *a*:

$1.275 \times 1.265 \times 3.14 = 5.065$, or 5 1/16 inches, nearly;

for gear *b*:

$3.225 \times 0.791 \times 3.14 = 8.010$, or 8 3/32 inches, nearly.

The lead of the helix must be, in general, the adjustable quantity in any spiral gear calculation. If special cutters are to be made, the lead of the helix may be determined arbitrarily from those given in the milling machine table; this will, however, probably necessitate a cutter of fractional pitch. On the other hand, by using stock cutters and varying the center distance slightly, we might find a combination which would give us for one gear a lead found in the milling machine table, but it would only be chance that would make the lead for the helix in the mating gear also of standard length. It is then generally better to calculate the milling machine change gears according to the usual methods to suit odd leads, rather than to adapt the other conditions to suit an even lead. It will be found in practice that the

lead of the helix may be varied somewhat from that calculated without seriously affecting the efficiency of the gears.

The remaining calculations relating to the proportions of the teeth do not vary from those for spur gears and are here set down for the sake of completeness only.

The addendum of a standard gear is found by Rule 7:

For gears *a* and *b* this will give:

$$1 \div 12 = 0.0833 \text{ inch.}$$

The whole depth of the tooth is found by Rule 8:

This gives for gears *a* and *b*:

$$2.157 \div 12 = 0.1797 \text{ inch.}$$

The thickness of the tooth is found by Rule 9:

For gears *a* and *b* of our problem this gives:

$$1.571 \div 12 = 0.1309 \text{ inch.}$$

This completes all the calculations required to give the essential data for making our first pair of helical gears. To illustrate the variety of conditions for which these problems may be solved, the other cases will be worked out somewhat differently. In the case just considered no allowance was made for possible conditions which might have limited the dimensions of the gears, and the problem was solved for what might be considered good general practice. Gear *a*, however, might have been too small to put on the shaft on which it was intended to go, while gear *b* might have been too large to enter the space available for it. If, as we may assume, these gears are intended to drive the camshaft of a gas engine, the solution would probably be unsatisfactory. Case No. 2 will therefore be solved for a center distance of 3⅜ inches as required, but the two gears will be made of about equal diameter. Fig. 7 shows the preliminary graphical solution of this problem, the reference letters in all cases being the same as in Fig. 6. With a 10-pitch cutter, if this suited the judgment of the designer, 15 teeth in gear *a* and 30 teeth in gear *b* would require that the point of intersection on the ratio line A E be located at O where B O equals the equivalent diameter of gear *a*, which equals 1½ inch, while B' O equals the equivalent diameter of gear *b*, or 3 inches, both calculated in accordance with Rule 11. The required condition now is that $D\,O$ be approximated to $D'\,O$; that is to say, that the pitch diameters of the two gears be about equal. After continued trial it will be found impossible to locate O, using a cutter of standard diametral pitch, so that $D\,O$ and $D'\,O$ shall be equal, and at the same time have $D\,D'$ equal to twice the required center distance, which is $2 \times 3\frac{3}{8}$ inches or 6¾ inches. If this center distance could be varied slightly without harm, B D could be taken as equal to A B; then it would be found that a line drawn from D through O to D', though giving a somewhat shortened center distance, would make two gears of exactly the same pitch diameter.

Drawing line $D\,O\,D'$, however, as first described to suit the conditions of the problem, and measuring it for a preliminary solution the following results are obtained: The tooth angle of gear a = angle

$B\,O\,D = 63° \ 45'$; and the tooth angle of gear B = angle $B'\,O\,D' = 90° - 63° \ 45' = 26° \ 15'$, according to Rule 1. Performing the operations indicated in Rule 12 to correct these angles, it is found that when the tooth angle of gear a is 63° 54′, and that for gear b is 26° 6′, the equation of Rule 12 becomes:

$$3 + (15 \times 2.04125) = 6.75 \times 0.89803$$
$$6.06187 = 6.06170$$

which is near enough for all practical purposes. The other dimensions

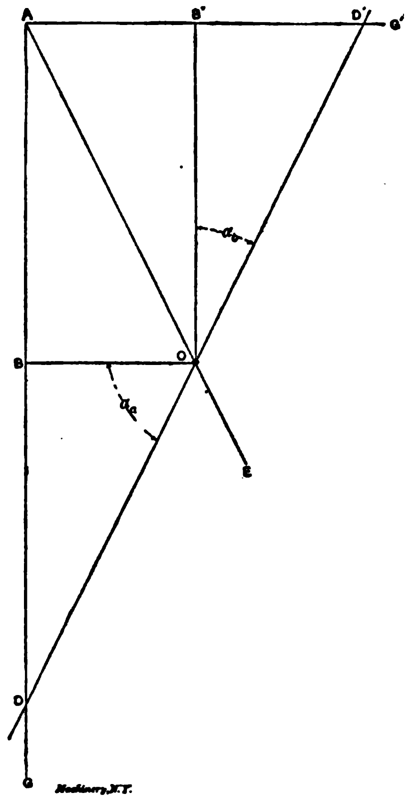


Fig. 7. Solution of Problem No. 2 for Equal Diameters

are easily obtained as before by using the remaining rules.

To still further illustrate the flexibility of the helical gear problem, the third case, for a center distance of 4 inches, will be solved in a third way. It is shown in MacCord's "Kinematics" that to give the least amount of sliding friction between the teeth of a pair of mating helical gears, the angles should be so proportioned that, in our diagrams, line $D\,D'$ will be approximately at right angles to ratio line $A\,E$. On the other hand, to give the least end thrust against the bearings, line $D\,D'$ should make an angle of 45° with the margin lines $A\,G$ and $A\,G'$, in

the case of gears with axes at an angle of 90°, as are the ones being considered. The first example worked out in detail was solved in accordance with "good practice," and line *D D'* was located about one-half way between the two positions just described, thus giving in some measure the advantage of a comparative absence of sliding friction, combined with as small degree of end thrust as is practicable. To illustrate some of the peculiarities of the problem, Case 3 will now be solved to give the minimum amount of sliding friction, neglecting entirely the end thrust, which is considered to be taken up by ball thrust bearings or some equally efficient device. On trial it will be found that, with the same number of teeth in the gear and with the same pitch as in Case 2, giving in Fig. 8, *B O*, the equivalent diameter of gear *a*, a value of 1½ inch, and *B' O*, the equivalent diameter of gear *b*, a value of 3 inches, as in Fig. 7, line *D D'* which is equal to twice

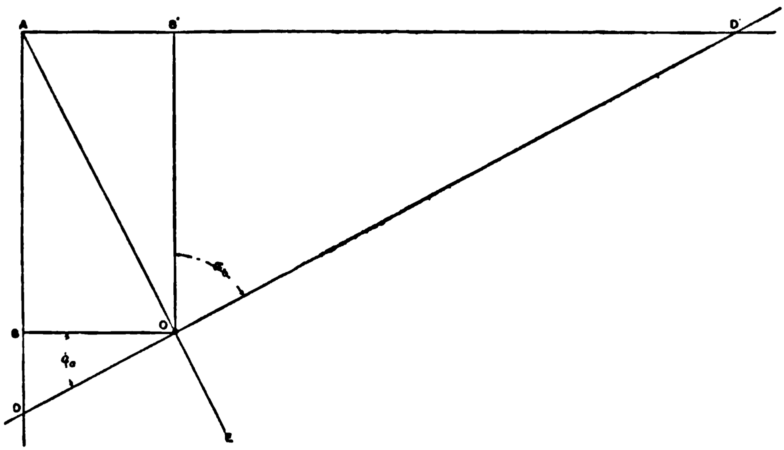


Fig. 8. Solution of Problem No. 3 for Minimum Sliding Friction

the center distance, or 8 inches, can then lie at an angle of about 90° with *A E*, thus meeting the condition required as to sliding friction. Thus this diagram, while relating to gears having the same pitch and number of teeth as Fig. 7, yet has an entirely different appearance, and gives different tooth angles and center distances, solving the problem as it does for the least sliding friction instead of for equal diameters of gears.

Measuring the diagram as accurately as may be, the following results are obtained: Tooth angle of gear $a = BOD = 28°$; tooth angle of gear $b =$ angle $B'OD' = 90° - 28° = 62°$. This is the preliminary solution. After accurately working it out by the process before described, we have as a final solution, tooth angle of gear $a = 28° 28'$; tooth angle of gear $b = 61° 32'$. From this the remaining data can be calculated.

For designers who feel themselves skillful enough to solve such problems as these graphically without reference to calculations, the diagram may be used for the final solution. The variation between the results

obtained graphically and those obtained in the more accurate mathematical solution is a measure of the skill of the draftsman as a graphical mathematician. The method is simple enough to be readily copied in a note book or carried in the head. If the graphical method is to be used entirely, it will be best not to trust to the cross-section paper, which may not be accurately ruled; instead skeleton diagrams like those shown in Figs. 6, 7 and 8 may be drawn. For rough solutions, however, to be afterward mathematically corrected, as in the examples considered in this chapter, good cross-section paper is accurate enough. It permits of solving a problem without drawing a line. Point O may be located by reading the graduations; a pin inserted here may be used as a stop for the rule, from which the diameter and center distance are read directly; dividing AD, read from the paper, by DD', read from the rule, will give the sine of the tooth angle of the gear a.

Formulas for Spiral Gearing

For sensible people, who prefer their rules to be embodied in formulas, the appended list has been prepared, using the following reference letters, which agree in general with the nomenclature of the Brown & Sharpe gear books.

N_a = number of teeth in gear a,
N_b = number of teeth in gear b,
P_n = normal diametral pitch or pitch of cutter,
γ = center angle,
α = angle of tooth with axis,
D = pitch diameter,
C = center distance,
N' = number of teeth for which to select cutter,
L = lead of tooth helix,
S = addendum,
W = whole depth of tooth,
T_n = normal thickness of tooth at pitch line,
O = outside diameter.

Where subscript letters $_a$ and $_b$ are used, reference is made to gears a and b, as for instance, "N_a" and "N_b," where the letter N refers to the number of teeth in gears a and b, respectively, of a pair of gears a and b.

$$\gamma = \alpha_a + \alpha_b \tag{1}$$

$$D = \frac{N}{P_n \cos \alpha} \tag{2}$$

$$C = \frac{D_a + D_b}{2} \tag{3}$$

$$N_b + (N_a \times \tan \alpha_a) = 2\,C\,P_n \times \sin \alpha_a \tag{4}$$

$$N' = \frac{N}{(\cos \alpha)^3} \tag{5}$$

$$L = \pi D \times \cot \alpha \tag{6}$$

$$S = \frac{1}{P_n} \tag{7}$$

$$W = \frac{2.157}{P_n} \tag{8}$$

$$T_n = \frac{1.571}{P_n} \tag{9}$$

$$O = D + 2S \tag{10}$$

Examples of Spiral Gear Problems*

A number of examples will be given in the following, which can be solved by simple modifications of the methods outlined for problems of Class 2. The same reference letters are used as before.

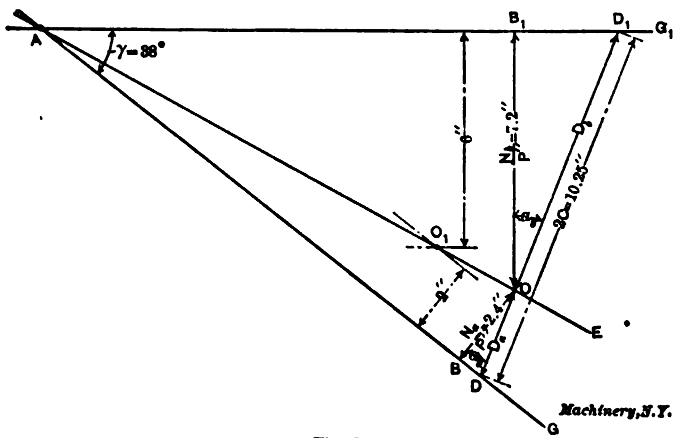


Fig. 9

Example 1.—Find the essential dimensions for a pair of spiral gears, velocity ratio 3 to 1, center distance between shafts 5⅛ inches, angle between shafts 38 degrees.

First obtain a preliminary solution by the diagram shown in Fig. 9. Draw lines AG and AG_1 making an angle γ with each other equal to 38 degrees, the angle between the axes. Locate the ratio line AE by finding any point such as O_1 between AG and AG_1, that is distant from each of them in the same ratio as that desired for the gearing. In the case shown, it is 6 inches from AG_1 and 2 inches from AG, which is in the ratio of 3 to 1 as required. Through O_1 draw line AE which may be called the ratio line. Select a trial number of teeth and pitch of cutter for the two gears, such, for instance, as 36 teeth for the gear and 12 for the pinion, and with 5 diametral pitch of the cutter. The diameter of a spur gear of the same pitch and number of teeth as the gear would be $36 \div 5 = 7.2$ inches. Find the point O

* MACHINERY, December, 1908.

on AE, which is 7.2 inches from AG_1. This point will be 2.4 inches from AG, if AE is drawn correctly.

Now apply a scale to the diagram, with the edge passing through O and with the zero mark on line AG, shifting it to different positions until one is found in which the distance across from one line to another (DD_1 in the figure) is equal to twice the center distance, or 10.25 inches. If a position of the rule cannot be found which will give this distance between lines AG and AG_1, new assumptions as to number of teeth and diametral pitch of the gear and pinion must be made, which will bring point O in a location where line DD_1 may be properly laid out. DD_1 being drawn, the problem is solved graphically. The tooth angle of the gear is B_1OD_1, or a_b, while that of the pinion

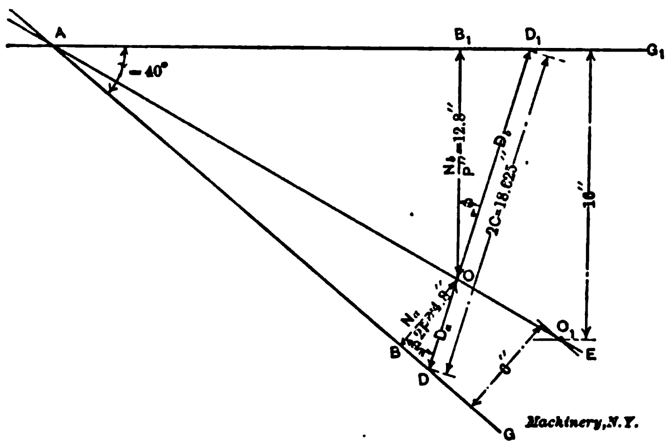


Fig. 10

is BOD, or a_a. OD_1 will be the pitch diameter of the gear, and OD the pitch diameter of the pinion.

To obtain the dimensions more accurately than can be done by the graphical process, the pitch diameters should be figured from the tooth angles we have just found. To do this, divide the dimensions OB_1 and OB for gear and pinion, by the cosine of the tooth angles found for them. If they measure on the diagram, for instance, 21 degrees 50 minutes and 16 degrees 10 minutes respectively (note that the sum of a_a and a_b must equal γ), the calculation will be as follows:

$$\begin{aligned} 7.2 \div 0.92827 &= 7.7563 = D_b \\ 2.4 \div 0.96046 &= 2.4988 = D_a \\ \hline &10.2551 = 2C \end{aligned}$$

The value we thus get, 10.2551 inches, for twice the center distance, is somewhat larger than the required value, 10.250 inches. We have now to assume other values for a_a and a_b, until we find those which give pitch diameters whose sum equals twice the center distance. Assume, for instance, that $a_b = 21$ degrees 43 minutes, then $a_a = 38$ de-

grees — 21 degrees 43 minutes = 16 degrees 17 minutes. We now have:

$$7.2 \div 0.92902 = 7.7501 = D_b$$
$$2.4 \div 0.95989 = 2.5003 = D_a$$
$$\overline{10.2504 = 2\,C}$$

This value for twice the center distance is so near that required that we may consider the problem as solved. The other dimensions for the outside diameter, lead, etc., may be obtained as for spiral gears at right angles, and as described in the previous part of this chapter.

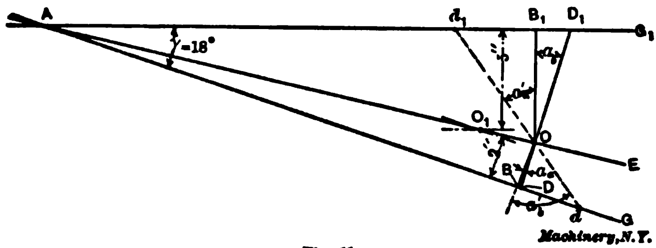


Fig. 11

Example 2.—Find the essential dimensions of a pair of spiral gears, velocity ratio 8 to 3, center distance between shafts 9 5/16 inches, angle between shafts 40 degrees.

The diagram for solving this problem is shown in Fig. 10. The axis lines $A\,G_1$ and $A\,G$ are drawn as before and the ratio line $A\,E$ is drawn in the ratio of 8 to 3, or 16 to 6, by the same method as just described. A point O is found having a location corresponding to 64 teeth and 5 pitch for the gear, and 24 teeth for the pinion. This gives distance $O\,B_1 = 12.8$ inches, and $O\,B = 4.8$ inches, by which position O is so located that a line $D\,D_1$ can be drawn through it at a convenient angle,

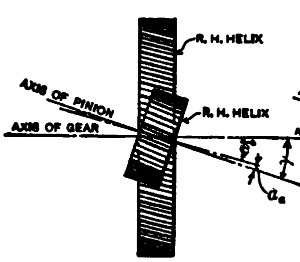


Fig. 12

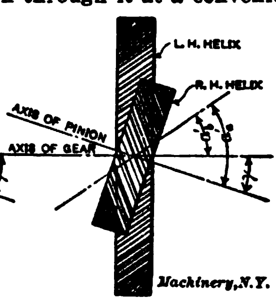


Fig. 13

and with a length equal to twice the center distance, or 18.625 inches. We measure the angle for a preliminary graphical solution as before, and then by trial find the final solution as follows, in which angle a_b is 17 degrees 45 minutes, and a_a is 22 degrees 15 minutes'

$$12.8 \div 0.95240 = 13.4397 = D_b$$
$$4.8 \div 0.92554 = 5.1862 = D_a$$
$$\overline{18.6259 = 2\,C}$$

This gives the value of twice the center distance near enough for gears of this size.

Example 3.—Find the essential dimensions for a pair of spiral gears, velocity ratio 5 to 2, center distance between shafts 4 1/16 inches, angle of shafts 18 degrees.

The diagram for solving this problem is shown in Fig. 11. The axis lines AG_1 and AG are drawn as before, and the ratio line AE is drawn in the ratio of 5 to 2, by the same method as just described. A point O is found having a location corresponding to 45 teeth and 8 pitch for the gear, and 18 teeth for the pinion. This gives distance $OB_1 = 5.625$ inches, and $OB = 2.250$ inches, in which position O is so located that line DD_1 can be drawn through it at a convenient angle, and with a length equal to twice the center distance, or 8.125 inches. We measure the angles for a preliminary mathematical solution as before, and then by trial find the final solution as follows, in which angle α_b is 16 degrees 45 minutes and α_a is 1 degree 15 minutes:

$$5.625 \div 0.95757 = 5.8742 = D_b$$
$$2.250 \div 0.99976 = 2.2505 = D_a$$
$$\overline{8.1247} = 2C$$

It is often a matter of great difficulty, when the center angle γ is as small as in this case, to find a location for point O such that standard cutters can be used, and that line DD_1 can be drawn of the proper length through O without bringing D to the left of B, or D_1 to the left of B_1. It will be noticed in this case that to make the center distance come right, angle α_a had to be made very small, so that the pinion is practically a spur gear. In some cases, to get the proper center distance, it may be necessary to so draw line DD_1 that one of the tooth angles is measured on the left side of BO or B_1O. Such a case, for instance, is shown in the position of d_1Od. When a line has to be drawn like this, the tooth angles α'_a and α'_b are opposite in inclination, instead of having them, as usual, either both right hand or both left hand. In Fig. 12 are shown gears drawn in accordance with the location of line DD_1 of Fig. 11, while Fig. 13 shows a pair drawn in accordance with dd_1 of the same diagram, which will illustrate the state of affairs met with in cases of this kind. This expedient of making one spiral gear right-hand and one left-hand should never be resorted to except in case of extreme necessity, as the construction involves a very wasteful amount of friction from the sliding of the teeth on each other as the gears revolve.

Demonstration of Grant's Formula

As already mentioned, the number of teeth for which the cutter should be selected for cutting a helical gear, is found to be the formula

$$N' = \frac{N}{\cos^3 \alpha}$$

in which N' = number of teeth for which cutter is selected,
N = actual number of teeth in helical gear,
α = angle of tooth with axis.

Note that $\cos^3 \alpha$ is equivalent to $(\cos \alpha)^3$.

A demonstration of this formula was presented by Mr. H. W. Henes in MACHINERY, April, 1908. This demonstration is as follows:

Let P_n be the perpendicular distance between two consecutive teeth on the spiral gear, and let D_1 be the diameter of the spiral gear. Let the gear be represented as in Fig. 14, and pass a plane through it perpendicular to the direction of the teeth. The section will be an ellipse as shown in $CEDF$. Designate the semi-major and semi-minor axes by a and b respectively.

Now N' is the number of teeth which a spur gear would have if its

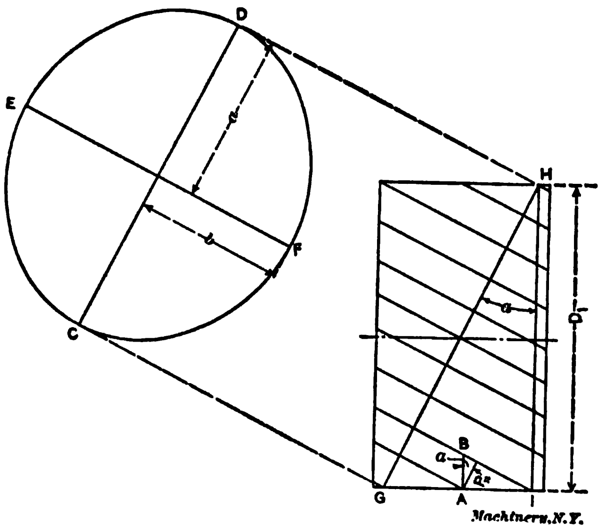


Fig. 14. Diagram for Deriving the Formula for Determining Spur Gear Cutter for Cutting Spiral Gears

radius were equal to the radius of curvature of the ellipse at E. Therefore, it is required to determine the radius of this curvature of the ellipse. This is done as follows:

From the figure we have:

$$2\,b = \text{axis}\ EF = D_1 \tag{11}$$

$$2\,a = \text{axis}\ CD = GH = \frac{HI}{\cos\alpha} = \frac{D_1}{\cos\alpha} \tag{12}$$

From (11) and (12) we have for a and b,

$$b = \frac{D_1}{2} \tag{13}$$

$$a = \frac{D_1}{2\cos\alpha} \tag{14}$$

It is known, and shown by the methods of calculus, that the minimum curvature of an ellipse, that is, the curvature at E or F, equals

$\frac{b}{a^2}$. Taking the values of a and b found in (13) and (14), we have the curvature at E:

$$\text{Curvature} = \frac{b}{a^2} = \frac{\frac{D_1}{2}}{\frac{D_1^2}{4\cos^2 a}} = \frac{4 D_1 \cos^2 a}{2 D_1^2} = \frac{2\cos^2 a}{D_1} \tag{15}$$

It is also shown in calculus that the *curvature* is equal to $\frac{1}{R}$, where R is the *radius of curvature* at the point E. Therefore from (15) we have:

$$\frac{1}{R} = \frac{2\cos^2 a}{D_1}, \text{ whence } R = \frac{D_1}{2\cos^2 a} \tag{16}$$

Formula (16) can also be arrived at directly, without reference to the minimum curvature of the ellipse, by introducing the formula for the radius of curvature in the first place. The curvature is simply the reciprocal value of the radius of curvature, and is only a comparative means of measurement. The radius of curvature of an ellipse at the end of its short axis is $\frac{a^2}{b}$, from which formula (16) may be derived directly by introducing the values of a and b from equations (13) and (14).

Having now found the radius of curvature of the ellipse at E, we proceed to find the number of teeth which a spur gear of that radius would have. From Fig. 14 we have:

$$A\,B = \frac{P_n}{\cos a} \tag{17}$$

Now, if $A\,B$ be multiplied by the number of teeth of the spiral gear, we shall obtain a quantity equal to the circumference of the gear; that is:

$$A\,B \times N = \pi D_1, \text{ and since } A\,B = \frac{P_n}{\cos a} \text{ from (17)}$$

$$\frac{P_n}{\cos a} \times N = \pi D_1 \tag{18}$$

Since N' is the number of teeth which a spur gear of radius R would have, then,

$$N' = \frac{2\pi R}{P_n} \tag{19}$$

In equation (19) the numerator of the fraction is the circumference of the spur gear whose radius is R, and the denominator is the circular pitch corresponding to the cutter.

From equation (16) we have:

$$R = \frac{D_1}{2 \cos^2 \alpha}$$

Substituting this value of R in (19), we have:

$$N' = \frac{2 \pi D_1}{P_n \times 2 \cos^2 \alpha} \qquad (20)$$

From equation (18) we have:

$$D_1 = \frac{N P_n}{\pi \cos \alpha} \qquad (21)$$

Substitute this value of D_1 in equation (20) and we have:

$$N' = \frac{2 \pi N P_n}{2 P_n \pi \cos^3 \alpha}$$

or

$$N' = \frac{N}{\cos^3 \alpha} \qquad (22)$$

Since N is the number of teeth in our spiral gear and N' is the number of teeth in a spur gear which has the same radius as the radius of curvature of the helix above referred to, this is the equivalent of saying that the cutter to be used should be correct for a number of teeth which can be obtained by dividing the actual number of teeth in the gear by the cube of the cosine of the tooth angle. Since the cosine of angle is always less than unity, its cube will be still less, so N' is certain to be greater than N, which will account for the fact that spiral gears of less than 12 teeth can be cut with the standard cutters.

CHAPTER II

DIAGRAMS FOR DESIGNING SPIRAL GEARS*

Great difficulties are usually experienced in designing spiral gears, and these difficulties are greatly accentuated when one has to design them for two shafts whose center distance cannot be altered to suit the gears, and also when the angle between the shafts is not a right angle, and the speed ratio is not equal. The general practice is to work out the gears by lengthy mathematics, and should the answer not come out as desired, then a new trial is made, varying either one or the other factor, until the angles and diameters are correct. This method of "cut and try" entails a great deal of work and waste of time. The following method, together with the diagrams used with it, will remove some of the difficulties, and enable one to arrive at the data required in a very short time. The method adopted is graphical, but the results may be checked by simple figuring.

As the pitch diameter, spiral angle, and circular pitch are interdependent, they cannot be considered as a starting point in solving the problem, because they are not known. The starting point, therefore, must be the speed ratio, and some idea of the strength required, together with the center distance. These factors, as a rule, can easily be ascertained. As it is common usage to employ ordinary spur gear cutters of regular diametral pitches for cutting spiral gears, the normal pitch, or distance from one tooth to the next measured at right angles to the tooth, must be the same as the pitch of a spur gear for which the cutter to be used is intended; therefore the corresponding diametral pitch and the speed ratio must be the initial data, all others being obtained afterwards.

Three diagrams are given for the graphical solution of spiral gears. The diagram in Fig. 15 shows the relation between the quotient of number of teeth ÷ diametral pitch, spiral angles, and pitch diameters. The quotient $\frac{\text{number of teeth}}{\text{diametral pitch}}$ is commonly termed "equivalent diameter," and will be so referred to in the following.

The diagram in Fig. 16 shows the relation between the diametral pitch, the number of teeth, and the equivalent diameter. Finally, the diagram in Fig. 17 shows the relation between the pitch diameter, the spiral angle, and the lead of the helix. We will now proceed to give some typical examples illustrating the use of the diagrams.

Example 1. Given a gear having 24 teeth, 6 diametral pitch, and a spiral angle of 40 degrees. Find the pitch diameter.

First obtain the value of the ratio, number of teeth ÷ diametral pitch, which, in this case, can be obtained without referring to dia-

* MACHINERY, October, 1908.

gram Fig. 16, being simply $24 \div 6 = 4$. Locate 4 on the horizontal line in diagram Fig. 15, and project vertically until the line from figure 4 intersects the line for 40 degrees spiral angle. Then follow the circular arc from this point, either to the right or downward, reading off 5.22 on the corresponding scale, this being the pitch diameter. Should the diameter be required accurately, we can figure it by the formula:

$$\text{Pitch diameter} = \frac{\text{No. of teeth}}{\text{Diametral pitch}} \times \frac{1}{\text{cos spiral angle}}$$

$$= 4 \times \frac{1}{\text{cos 40 deg.}} = 5.222 \text{ inches.}$$

This also gives a check of the result obtained by means of the dia-

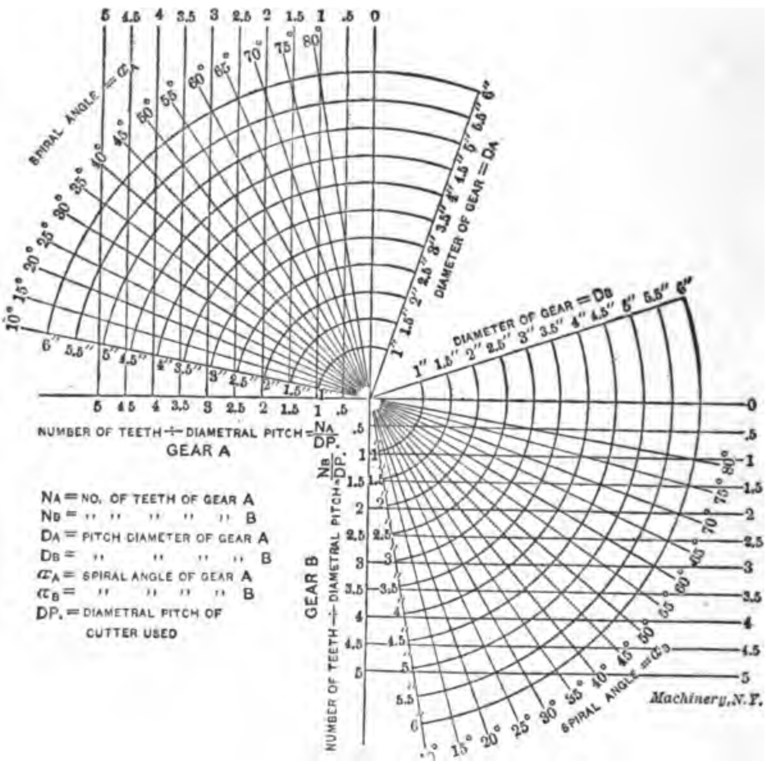


Fig. 15. Diagram of Relation between Number of Teeth, Diametral Pitch, Spiral Angles and Pitch Diameters

gram. The lead of the helix is now obtained from Fig. 17, by projecting the pitch diameter 5.22 horizontally to the radial line for the spiral angle, and then, following the vertical line to the lead scale at the bottom of the diagram, we find, in this case, a lead of 19.6 inches. Of course, the outside diameter of the blank would be 5.222 +

$2 \times 1/6 = 5.555$ inches, which is the pitch diameter + 2 times the addendum.

Example 2. Required two gears which are to be equal in all respects, the diametral pitch being 8, and the centers to be approximately 4 inches apart.

As the centers are not fixed, the gears in this case may be made with 45 degrees spiral angle, and the center distance may be slightly adjusted to suit the pitch diameters. Referring to Fig. 15, follow the circular arc from diameter of gear = 4 inches, until it intersects the radial line for 45 degrees spiral angle; then follow the vertical line down to the scale of the ratio between the number of teeth and diam-

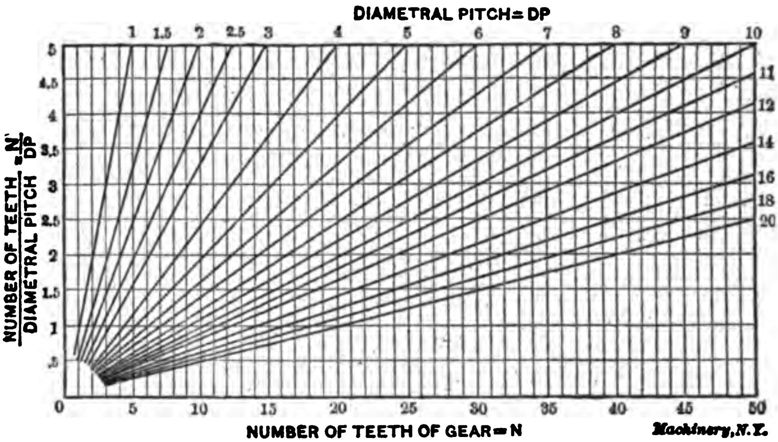


Fig. 16. Relation between Diametral Pitch, Number of Teeth, and Quotient of Number of Teeth divided by Diametral Pitch

etral pitch, which is found to be 2.82. Then, from Fig. 16, we find that with this ratio and 8 diametral pitch, the number of teeth is not a whole number, but the nearest number is 23, giving a ratio of 2.875 instead of 2.82, which, by reversing the process and referring to diagram Fig. 15, gives a pitch diameter of 4.07 inches. These results may be checked as follows:

$$\text{Pitch diameter} = \frac{\text{No. of teeth}}{\text{Diametral pitch}} \times \frac{1}{\cos 45 \text{ deg.}}$$

$$= 2.875 \times \frac{1}{0.707} = 4.07 \text{ inches.}$$

The outside diameter is $4.07 + 2 \times 0.125 = 4.32$. The lead, as obtained from diagram Fig. 17, in the same way as in Example 1, is 12.79 inches.

Example 3. Required a pair of spiral gears having a normal pitch corresponding to 10 diametral pitch, having a given center distance of 2½ inches approximately, the sum of the spiral angles being 90 degrees, and the speed ratio equal to 5 to 1.

In this case both portions of diagram Fig. 15 are used, the upper part

being employed for one gear and the lower part for the other, the easiest way being to get a strip of paper with two lines marked on its edge 5 inches (twice the center distance) apart, drawn to the same scale as the diagram. Move this strip of paper on the diagram (so that the edge of the strip passes through the center), as indicated at *A*, Fig. 18, until the lines marked coincide with the points where the ratio of the equivalent diameters equals 5 to 1, and then determine from Fig. 16 that these diameters also give whole numbers of teeth with 10 diametral pitch. We find that 0.5 and 2.5 at 78 degrees and 12 degrees are two such positions, and also 0.6 and 3.0 at 70 degrees and 20 degrees. If we use the latter values, we will have 6 teeth and 30 teeth at 70 and 20 degrees angle, respectively. The exact di-

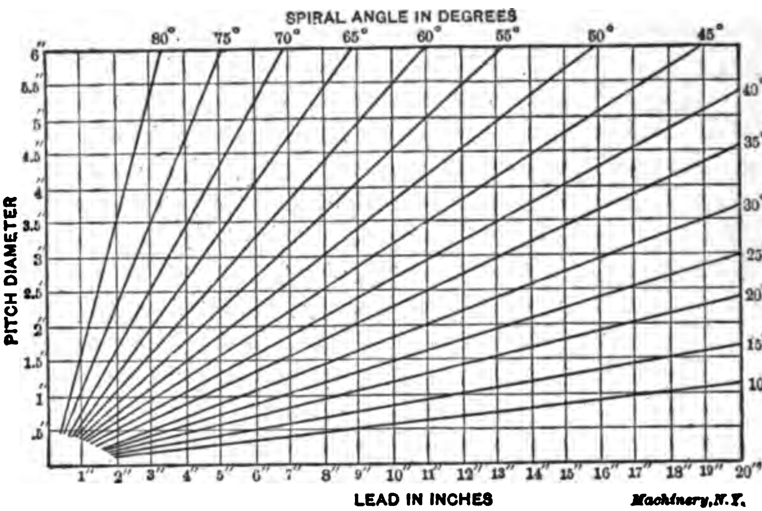


Fig. 17. Relation between Pitch Diameter, Spiral Angle and Lead of Helix

ameters can now be determined, as in our previous problem, and are 1.75 and 3.19 inches, respectively, the outside diameters being 0.2 inch larger, or 1.95 and 3.39 inches, respectively. This gives the center distances 2.47. These values can now be obtained from the formulas as before.

Example 4. Required a pair of spiral gears, having a fixed center distance of 4.5 inches, running at equal speeds, the diametral pitch being 7. The method of procedure is similar to that of the last example, using a strip of paper having a distance of 9 inches marked on the edge in the proper scale, as indicated at *B* in Fig. 18. At about 40 degrees spiral angle we find in Fig. 15 the ratio of number of teeth to diametral pitch to equal 3.14. This ratio must be adjusted on diagram Fig. 16, as previously shown, so as to enable one to get a whole number of teeth with 7 diametral pitch, this number being in this case 22. The ratio is then 3.143, and following from this in Fig. 15 to the 40-degree line, one obtains a pitch diameter of about 4.1 inches for one gear, and at 50 degrees about 4.9 inches for the other.

The spiral angles should now be carefully checked mathematically as follows:

$$\cos \text{spiral angle (first gear)} = 3.143 \times \frac{1}{4.1} = 0.766;\ \text{spiral angle} = 40 \text{ deg.}$$

$$\cos \text{spiral angle (second gear)} = 3.143 \times \frac{1}{4.9} = 0.642;$$

$$\text{spiral angle} = 50 \text{ deg., nearly.}$$

Now obtain the leads from diagram Fig. 17 in the same way as before, giving the leads of the gears 15.4 and 12.9 inches, respectively.

Example 5. Required a pair of spiral gears, the axes of which are at an angle of 120 degrees; center distance 4.125; the ratio of equivalent diameters should be as 2 to 3, and the diametral pitch equals 5.

We require first of all two numbers representing the equivalent diameters, these two numbers bearing the ratio to each other of 2 to 3, and giving a whole number of teeth with 5 diametral pitch. These

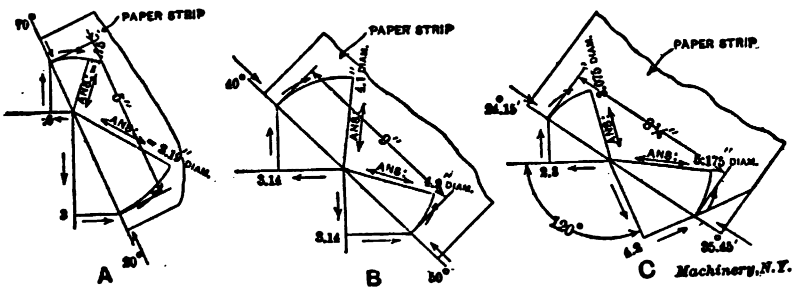
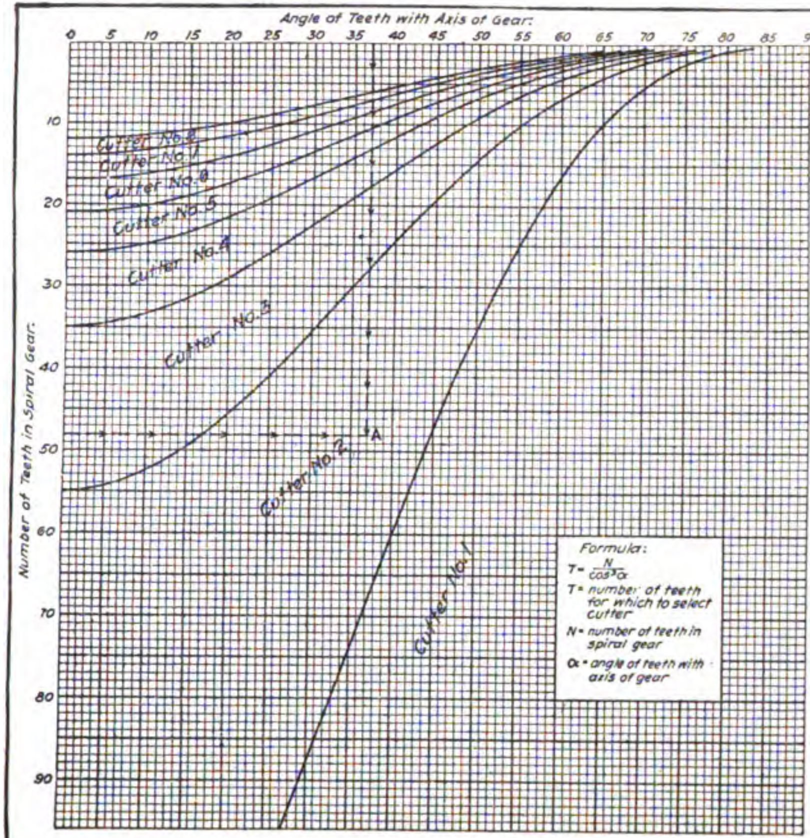


Fig 18. Separate Diagrams for the Solution of some of the Problems Presented

two numbers, when projected onto two spiral angle lines in a diagram made up as in Fig. 15, the sum of the angles of which equals 120 or 60 degrees, give two diameters whose sum equals the center distances multiplied by 2, or 8.25. In this case we cannot use both parts of the diagram Fig. 15, as it is made up for shafts at 90 degrees angle, and for this reason we must take the two readings from the same part of the diagram. The ratios 3 and 4.5 at 30 degrees give corresponding diameters of 3.5 and 5.2, the sum being 8.7. The ratios 2.8 and 4.2 giving 14 and 21 teeth at 25 and 35 degrees, respectively, have diameters of 3.1 and 5.15 (equals 8.25). From this we see that we must use 14 and 21 teeth and the ratios 2.8 and 4.2. The diameters and spiral angles can now be obtained graphically and more accurately in this manner:

Draw two radial lines, as shown at *C* in Fig. 18, at 120 degrees angle on a separate piece of paper, and lay off on these to same scale 2.8 and 4.2. From these points draw lines at right angles to the radial lines. It is now necessary to find the position of a line 8.25 inches long, terminating upon these lines, and passing through the center. A



The cutters used for milling spiral or helical gears are standard spur gear cutters, the number of a cutter and its pitch for a given case being defined by the angle (with axis) and normal pitch. This diagram gives the numbers of the cutters only, the pitch having been previously determined.

The selection of the cutter is fixed by the formula given in the lower right-hand corner of the diagram. The delimiting curves thereon were plotted by the formula, the area between the curves being the field of intersection of the combinations of angles and numbers of teeth covered by each designated cutter number.

For example, suppose the angle of the teeth of a gear is 37 degrees with its axis, and the number of teeth is 48. The point A, at which the horizontal line (representing the tooth number), and the vertical line (representing the angle) intersect, falls within the area marked "Cutter No. 2" Therefore, a No. 2 cutter is required to cut a 48-tooth spiral gear having the teeth at an angle of 37 degrees with its axis.

Fig. 19. Diagram for Determining Cutter to Use for Milling Spiral Gears

strip of paper is used in the same manner as before, and upon careful measuring of the respective distances from the center to the lines, one obtains the distances 3.075 and 5.175 inches, which represent the respective diameters, the sum being 8.25. The spiral angles are obtained by measuring or calculating as follows:

$$\cos \text{ spiral angle of first gear} = 2.8 \times \frac{1}{3.075} = 0.910;$$

$$\text{spiral angle} = 24 \text{ deg. } 15 \text{ min.}$$

$$\cos \text{ spiral angle of second gear} = 4.2 \times \frac{1}{5.175} = 0.812;$$

$$\text{spiral angle} = 35 \text{ deg. } 45 \text{ min.}$$

The above examples will show the careful student the manner of working out various problems as required, and if the directions are properly followed, this method will be found to be a great time-saver. It may be mentioned that it is advisable to keep the spiral angle as nearly equal in the two gears as possible in order to obtain the greatest efficiency of transmission. It should be noted that when diagrams of this type are to be used for practical calculation of spiral gears, they should be laid out in a much larger scale than is possible to show in these pages, and it would be advisable to lay out radial lines in Fig. 15 for every degree, and vertical and horizontal lines for every tenth of an inch, and circular arcs for equally fine subdivisions. The same is true of the diagrams in Figs. 16 and 17. In Fig. 16, horizontal lines should be laid out for every tenth of an inch, and vertical lines should be laid out for all whole numbers of teeth. In Fig. 17, the horizontal lines should be laid out for every tenth of an inch, vertical lines for at least every 0.2 of an inch, and radial lines for every degree. This diagram should also be laid out so that leads over 20 inches may be read off, as well as those below this figure.

In Fig. 19 is given a diagram for determining the cutter to use when milling the teeth of spiral gears. The instructions for the use of the diagram are given directly on the chart itself, so that no other explanation is necessary. This diagram was contributed to MACHINERY by Elmer G. Eberhardt, and appeared in the September, 1907, issue.

CHAPTER III

HERRINGBONE GEARS*

The following information on herringbone gearing is abstracted from a paper by Mr. Percy C. Day, of Milwaukee, Wis., read before the meeting of the American Society of Mechanical Engineers, under the auspices of the sub-committee on machine shop practice, at New York, December 5-8, 1911. This abstract was published in MACHINERY, January, 1912.

That the helical principle in toothed gearing is ideal from a theoretical point of view is well known. From a practical standpoint, so called "herringbone" gears have, however, been less satisfactory than straight-cut spur gears, because, until recently, no method was devised for producing them with the requisite speed and accuracy. Within the last six years, however, a method has been developed, in England, to a high degree of perfection. Herringbone gears made by this method are called Wuest gears, after the inventor. The distinction between these gears and those of the ordinary herringbone type is that the teeth of the former, instead of joining at a common apex at the center of the face, are stepped half the pitch apart and do not meet at all. This arrangement of the teeth does not affect the action of the gears, but it facilitates their commercial production.

Action of Spur Gearing

The aim of all designers of gearing is to transmit rotary motion from one axis to another in a perfectly even manner without variation of angular velocity. Let us consider the action of a straight spur pinion driving a gear. There are three distinct phases of engagement:

First phase: The root of the pinion tooth engages the point of the gear tooth.

Second phase: The teeth are engaged near the pitch line.

Third phase: The point of the pinion tooth engages the root of the gear tooth.

Let us assume that the teeth are accurately cut to involute form, so that if the pinion moves with even angular velocity it will produce corresponding evenness of motion in the gear; and also that the pinion has sufficient teeth to allow the engagement of successive teeth to overlap. At the beginning of the first phase, while the load is carried near the point of the gear tooth, that tooth is subjected to a maximum bending stress along its whole length. The portion of the pinion tooth near the root is sliding over the outer portion of the gear tooth; that is to say, two metallic surfaces of small area are sliding under heavy compression.

*MACHINERY, Engineering Edition, January, 1912.

The action during the second phase more nearly approaches ideal conditions. The teeth are engaged near their respective pitch lines and very little sliding takes place. During the third and final phase, the pinion tooth is subjected to a maximum bending stress, while the tooth surfaces again slide over each other, this time with the outer portion of the pinion tooth engaging the gear tooth near its root. The point to be noted is that while those portions of the mating teeth which are near the pitch lines transmit the load with rolling contact, those which are more remote have to transmit the same load with sliding contact. The inevitable result is that the points and roots of all the teeth tend to wear away more rapidly than the portions near the pitch lines.

It may be suggested that the sliding action can be eliminated by shortening the teeth so that they engage only the phase of rolling contact. This has been tried with a certain measure of success in the stub-toothed gear, but it cannot be carried far enough without curtailing the arc of contact so that continuity of engagement is lost.

Action of Herringbone Gears

Herringbone gears completely overcome all these difficulties, but only when they are accurately cut. If we take two exactly similar pinions with straight teeth and place them side by side on one shaft, with the teeth of one pinion set opposite the spaces of the other, then we have what is known as a stepped-tooth pinion. If this pinion is meshed with a composite gear made up in a similar manner, the action is modified so that there are always two phases of engagement taking place simultaneously. Such gears are commonly used for rolling mill work, because they stand up to heavy shocks better than the plain type. Still better action can be secured by assembling a number of narrow pinions with the last of the series one pitch in advance of the first and the others advanced by equal angular increments. As a practical proposition, however, gears made on these lines would be costly and difficult to produce.

The helical gear is the logical outcome of the stepped gear carried to its limit, and built up from infinitely thin laminations. Since the steps have merged into a helix, there must be a normal component of the tangential pressure on the teeth, producing end thrust on the shafts. To obviate end thrust, the helical teeth are made right-hand on one side and left-hand on the other. (See Fig. 20.) Such gears, with double helical teeth, are known as herringbone gears.

The fundamental principle of the action of herringbone teeth lies in the circumstance that *all phases of engagement take place simultaneously.* This holds good for every position of pinion and gear, provided only that the relationship between pitch, face width, and spiral angle is such as will insure a complete overlap of engagement. Since all phases of engagement occur together, it follows that the load is partly carried by tooth surfaces in sliding contact and partly by surfaces in rolling contact.

Those portions of the teeth farthest from the pitch line, which engage with sliding action, tend to wear away more rapidly than the portions nearest the pitch line. But the pitch line portion is always carrying part of the load, and the effect of wear on the ends of the teeth merely tends to throw more load on the center portions; in other words *there is a tendency to concentrate the load near the pitch lines.* The ends of the teeth, instead of wearing away to an ever-increasing extent from their original involute form, are relieved of some of the load from the moment that wear commences to take place. As soon as the load on these ends has been partially relieved and transferred to the middle portion, the wear becomes equalized all over the teeth and they do not tend to distort further from their original shape.

As the teeth keep their involute form, motion is transmitted from pinion to gear in an even manner, without jar, shock, or vibration. While herringbone teeth may not be intrinsically stronger than straight teeth, the elimination of shock renders them capable of transmitting heavier loads. Since all phases of engagement occur simultaneously, the transference of the load from one pinion tooth to the next takes place gradually instead of suddenly. This is the second principle of herringbone gearing, and may be termed *continuity of action.* In straight gears the continuity of action is a function of the number of teeth in the pinion. In herringbone gears continuity depends on the relationship between the face width and the number of teeth in the pinion. Pinions with as few as five teeth have been used with success by merely increasing the face width to suit such extreme conditions. This feature, which is peculiar to herringbone gears, has made practical the adoption of extremely high ratios of reduction hitherto considered impossible.

The third principle of herringbone gearing is that the bending stress on the teeth does not fluctuate from maximum to minimum as in straight gears, but remains always near the mean value. This feature is of special importance in rolling-mill driving and work of a similar nature.

To summarize the foregoing statements: The action of herringbone gears is continuous and smooth; there is no shock of transference from tooth to tooth; tne teeth do not wear out of shape; the bending action of the load on the teeth is less than with straight gearing and does not fluctuate to anything like the same extent; the gears work silently and without vibration; back-lash is absent; friction and mechanical losses are reduced to a minimum; herringbone gears can be used for higher ratios and greater velocities than any other kind.

It has been explained that the teeth of the Wuest gears are so designed that those on the right- and left-hand sides of the gears are stepped half a space apart, and do not meet at a common apex at the center of the face, as in the usual type of herringbone gear. It has often been argued that the ordinary herringbone tooth is stronger than the Wuest tooth, because the latter lacks the support given by the junction of the teeth at the center. This argument would be sound if

gear teeth were ever stressed to anywhere near their breaking point. But it has been found in practice that considerations of wear so far outweigh those of mere breaking strength that a gear which is designed to give reasonable service will carry anywhere from ten to twenty times the working load without fracture. A point of vastly greater importance is that the stepped form will wear more evenly under extreme loads than the ordinary type. The reason for this is shown in Figs 20 and 21. The resultant tooth pressure is always normal to the teeth and tends to bend them apart. The stepped form offers a uniform resistance along its whole length, carrying the load from end to end (Fig. 20). The teeth of ordinary herringbone gears tend

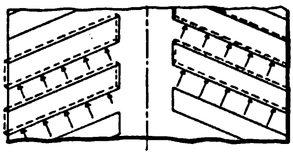


Fig. 20

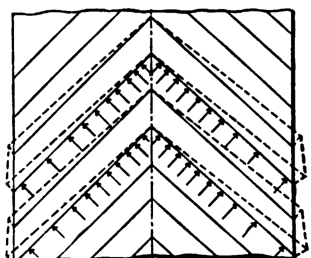


Fig. 21

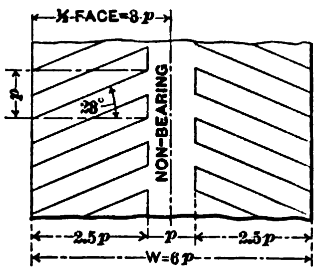


Fig. 22

Figs. 20 to 22. Diagrams showing Tooth Pressures and Angle Necessary for Continuity of Action

to separate more at the sides than near the supported center, causing the load to be concentrated toward the center (Fig. 21).

The standards which have been adopted for Wuest gears are the result of experience gained in Europe during the last six years. The spiral angle of the teeth is about 23 degrees with the axis. Since the nature of the action eliminates shock, it follows that the pitch required for given conditions will be much finer than would be chosen for spur gears. On the other hand, the face width will not be less, because there is as much necessity for wearing surface with one kind of tooth as with the other. Spur gears are usually made with a face width equal to three or four times the pitch. Herringbone gears may conveniently have a face width equal to six times the pitch, not because the width of this type need actually be greater, but by reason of the pitch being proportionately less.

Starting with a width equal to six times the pitch, and allowing one times the pitch as the non-bearing portion in the center, there remains

two and one-half times the pitch available for the teeth on each side. To insure continuity of engagement under all ordinary conditions, each tooth is inclined so as to cover an advance of one times the pitch within its length. The angle of 23 degrees satisfies this requirement (see Fig. 22).

The pressure angle which has been adopted for standard gears is 20 degrees. The teeth are shorter than the usual standards, because the high ratios used with these gears call for an average pinion diameter which is less than is used with straight spur gears for similar duty. The teeth are generated by hobs, and the short addendum combined with wide angle gives satisfactory tooth shapes, without undercutting of teeth on small pinions.

The dimensions proposed for an interchangeable system for these gears are as follows:

Tooth shape ....Involute
Pressure angle ....20 degrees
Spiral angle ....23 degrees

$$\text{Pitch diameter (20 teeth and over)} = \frac{\text{Number of teeth}}{\text{D.P.}}$$

$$\text{Blank diameter (20 teeth and over)} = \frac{\text{Number of teeth} + 1.6}{\text{D.P.}}$$

$$\text{Pitch diameter (under 20 teeth)} = \frac{0.95 \times \text{Number of teeth} + 1}{\text{D.P.}}$$

$$\text{Blank diameter (under 20 teeth)} = \frac{0.95 \times \text{No. of teeth} + 2.6}{\text{D.P.}}$$

Addendum .... $\frac{0.8}{\text{D.P.}}$

Dedendum .... $\frac{1.0}{\text{D.P.}}$

Full depth .... $\frac{1.8}{\text{D.P.}}$

Working depth .... $\frac{1.6}{\text{D.P.}}$

Standard face width for gears with pinions of not less than 25 teeth, 6 times circular pitch; face widths for high-ratio gears with small pinions, 6 to 12 times circular pitch.

When a pinion of less than 20 teeth is used with a standard gear, the center distance must be slightly increased to suit the enlargement of the pinion. If it is desired to keep the center distance to the standard dimensions, the gear diameter may be reduced by the amount of the enlargement given to the pinion. For example: If a pinion of 10 teeth, 5 diametral pitch is to mesh with a gear of 90 teeth at 10-inch centers,

$$\text{Pitch diameter of pinion} = \frac{0.95 \times 10 + 1}{5} = 2.1 \text{ inches.}$$

$$\text{Enlargement over standard pinion} = 0.1 \text{ inch,}$$

$$\text{Pitch diameter of standard gear} = \frac{90}{5} = 18.0 \text{ inches.}$$

$$\text{Reduced pitch diameter of gear} = 18.0 - 0.1 = 17.9 \text{ inches.}$$

$$\text{Center distance} = \frac{17.9 + 2.1}{2} = 10 \text{ inches.}$$

Power Transmitted by Herringbone Gears

The important factor in determining the proportions of the teeth is the relationship between pitch line velocity and the permissible

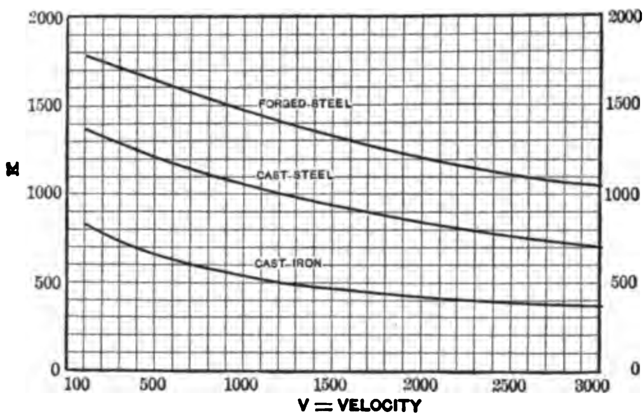


Fig 23. Shearing Stress with Relation to Pitch Line Velocity

specific tooth pressure; in other words, the total tooth pressure divided by the area of all the available simultaneous contact along the teeth. Theoretically, this contact has no area since it should consist of lines without breadth. Actually, an area exists, due to the elastic compression of the teeth in contact, in a similar way in which an area of contact exists between a car wheel and a rail. The area of contact is indeterminate, but the specific tooth pressure is proportional to the driving stress on the teeth.

In order to obtain a simple rule for finding the proper dimensions, the results of experience in the matter of safe working loads under given conditions have been reduced to a relationship between pitch line velocity and the shearing stress on the pitch line thickness of an imaginary straight tooth, assuming only one tooth in engagement at a time. The shearing stress is a measure of the specific tooth pressure, and the relationship referred to affords a convenient means of arriving at reliable dimensions. The curves, Fig. 23, give values of shearing stress K in pounds per square inch on the pitch line section of

an imaginary single tooth for corresponding pitch line velocities V in feet per minute. The values are entirely empirical, but they are based on the results of extended experience, and lead to dimensions which are safe and reliable. Different curves are given for different materials, and it is necessary to use that curve which corresponds to the lowest grade material of the combination. The dimensions of gears can be derived from the curves in the following manner:

H.P. = brake horsepower transmitted,
N = revolutions per minute,
D = pitch circle diameter, inches,
p = circular pitch in inches (use nearest diametral pitch),
W = total width of face, inches,
V = pitch line velocity, feet per minute,
P = total tooth pressure at pitch line, pounds,
K = stress factor (from curve).

Then

$$V = \frac{\pi D N}{12} \qquad P = \frac{\text{H.P.} \times 33{,}000}{V} \qquad P = \frac{p W K}{2}$$

$$P = 8\,p^2 K \left\{ \begin{array}{l} \text{in normal gears of moderate ratio, and face} \\ \text{width equivalent to six times the circ. pitch} \end{array} \right\}$$

$$p = \sqrt{\frac{P}{3K}}$$

For high ratio gears take $W = Rp$ (R = ratio) up to maximum of $W = 10p$.

$$p = \sqrt{\frac{2.5\,P}{R K}}$$

In normal gears it is safe to aim at pitch line velocities between 1000 and 2000 feet per minute, with 1500 feet as a fair average. If the pinion is to be fixed to a motor shaft without external support, the diameter must be greater than when it can be supported on both sides. Cast iron is preferable to cast steel for gears of large diameters and moderate power, but the latter will be found more economical for high tooth pressures. Pinions are usually made from steel forgings of 0.40 to 0.50 per cent carbon. Soft pinions should never be used for herringbone gears.

There are two special cases where the ordinary methods of calculation should not be used. Rolling-mill gears are subjected to stresses which are so far in excess of the average working load that it is necessary to consider carefully the strength of the teeth in regard to possible overloads. Extra high velocity gears, such as are used for steam turbines, require additional wearing surface and are characterized by extreme width of face combined with abnormally fine pitch.

CHAPTER IV

CALCULATING GEARS FOR GENERATING SPIRALS ON HOBBING MACHINES*

From time to time formulas have been developed for calculating the gears to be used for generating spiral gears. Those published in the past, however, have applied only to certain types of gear-hobbing machines. In the following a formula is given which is applicable to any type of gear-hobbing machine, and which is simpler to use than any formula so far published. In developing this formula, simple arithmetical expressions have been made use of, as far as possible, in order to make it especially useful to the practical man.

In order to clearly understand the use of any formula, it is necessary to know something of the principles involved. Fig. 24 shows a top view of a standard hobbing machine (the No. 3 Farwell) designed for cutting spur gears. Before dealing with the change gear ratios for spiral work, it will be well to have the methods for cutting spur gears firmly fixed in our minds. Assume the hob to be single threaded. It is evident that for each revolution of the hob, the gear being cut must move one tooth. Therefore, the hob revolves, for each revolution of the blank, as many times as there are teeth to be cut. To cut 44 teeth, we must gear the table to revolve once for every 44 revolutions of the hob.

The bevel gearing at *D*, Fig. 25, has a ratio of 3 to 1, the worm at *E* is double-threaded, and the worm-wheel *F* has 40 teeth. Hence the shaft *B* must revolve 3×44 times for each revolution of the table, and the worm shaft *C* must revolve 20 times for each revolution of the table. Hence we have:

$$\frac{\text{Revolutions of } B}{\text{Revolutions of } C} = \frac{3 \times 44}{20}$$

Inverting this ratio to get the change gear ratio required to obtain this result, we have:

$$\frac{20}{3 \times 44} = \frac{\text{Product of No. of teeth in driving gears}}{\text{Product of No. of teeth in driven gears}}$$

In the following formulas, we will designate the product of the number of teeth in the driving gears *P*, and the product of the number of teeth in the driven gears *p*.

Should we use a double-threaded or triple-threaded hob, the gear we are cutting must revolve two or three teeth for each revolution of the hob; in other words, the speed of the table is increased directly as the number of threads on the hob, so we must multiply the number

*MACHINERY, Engineering Edition, December, 1911.

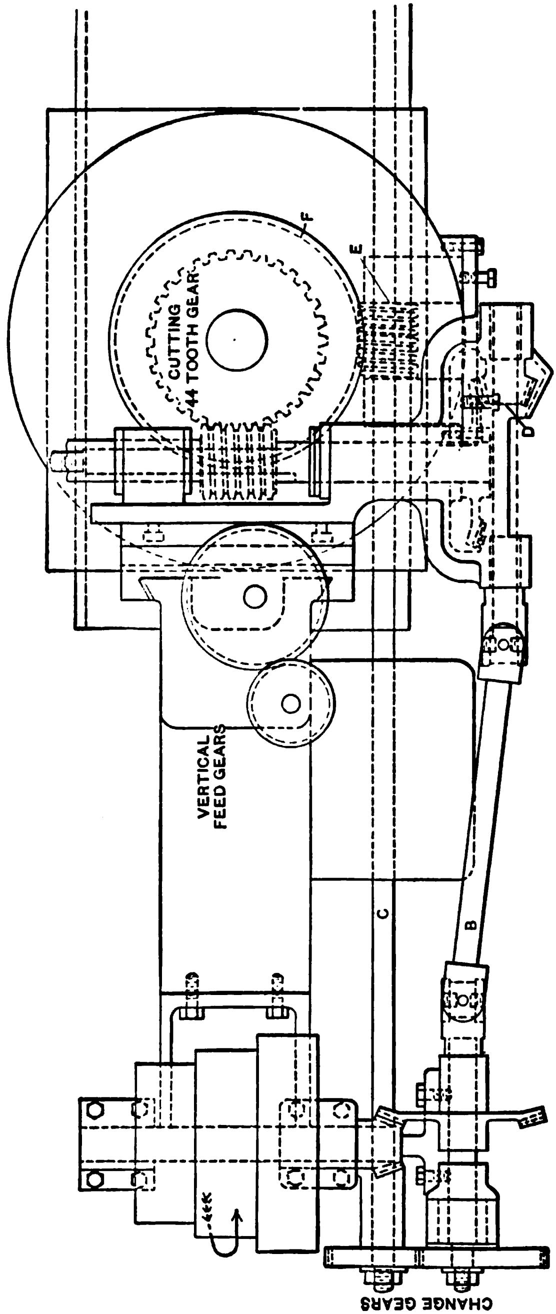


Fig. 24. Hob and Table Driving Arrangement of a No. 8 Farwell Gear-hobbing Machine

of teeth in the driving gears by the number of threads on the hob, giving us this formula:

$$\frac{20 \times \text{No. of threads on hob}}{3 \times \text{No. of teeth to be cut}} = \frac{P}{p}$$

A similar formula may be worked out in this way for any type of gear hobber.

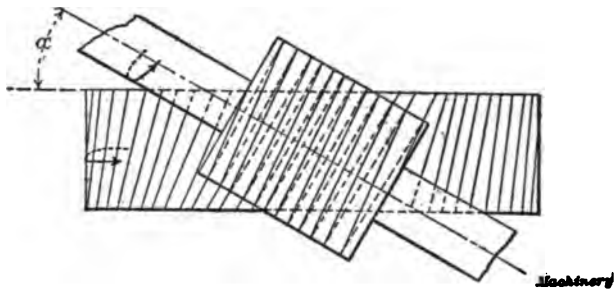


Fig. 25. Cutting a Right-hand Spiral Gear with a Right-hand Hob

Generating Spirals

For each revolution of the table, the head carrying the hob feeds down a certain distance across the face of the blank, this distance varying from 0.010 to 0.150 inch in common practice. To fully understand the following discussion, the action of the machine, as illustrated in Figs. 25 to 28, inclusive, should be noted. In Fig. 25 is shown the generation of a right-hand spiral gear with a right-hand hob; in Fig.

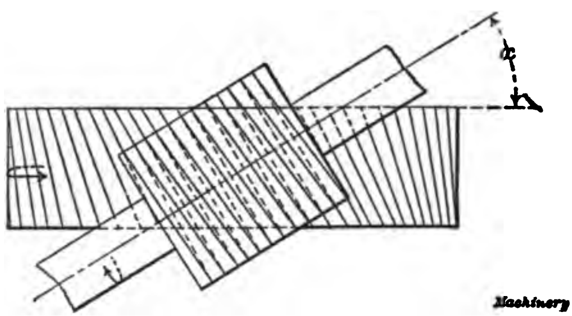


Fig. 26. Cutting a Left-hand Spiral Gear with a Right-hand Hob

26, a left-hand spiral gear with a right-hand hob; in Fig. 27, a left-hand spiral gear with a left-hand hob; and in Fig. 28, a right-hand spiral gear with a left-hand hob. In each of these illustrations the direction of rotation of the table is indicated by the arrow showing the direction of rotation of the gear being cut. The direction of rotation of the hob is also indicated by an arrow showing the direction of rotation of its shaft. In Figs. 25 and 27, where a gear is cut with

a hob of the same "hand," the angle a, as indicated, equals the difference between the tooth angle and the thread angle of the hob. In Figs. 26 and 28, where the gear and the hob are of different "hand," the angle a equals the sum of the tooth angle and the thread angle of the hob. After this preliminary introduction, we are ready to deal intelligently with the problem in hand.

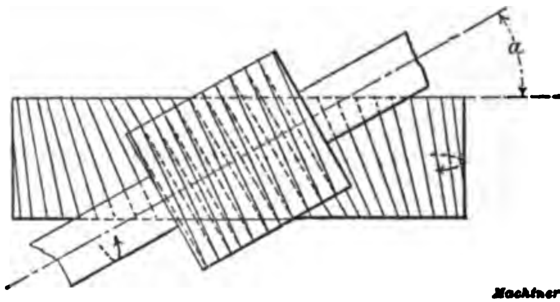


Fig. 27. Cutting a Left-hand Spiral Gear with a Left hand Hob

Assume the spiral gear shown in Fig. 29 to have sixty-four teeth. As indicated, the gear has a left-hand spiral and we will assume that it is cut with a left-hand hob. A single-threaded hob cutting a spiral gear would revolve sixty-four times for one revolution of the table; but since in this case the teeth are helical and the hob travels downward a certain distance, the position of the gear tooth must be advanced somewhat for every revolution with relation to the hob. In

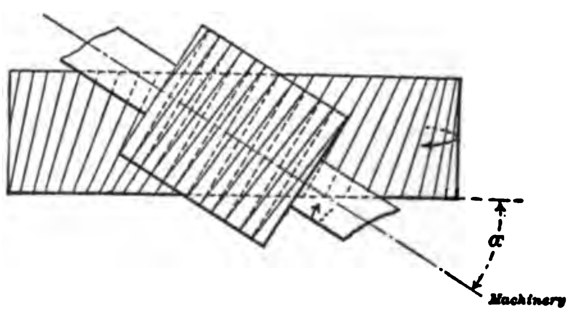


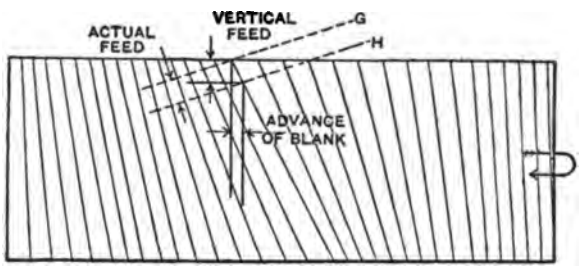
Fig. 28. Cutting a Right-hand Spiral Gear with a Left-hand Hob

other words, if the hob revolves sixty-four times, sixty-four teeth will have passed by, but the blank is not in the same position as at the beginning.

In Fig. 29, G represents the position of the hob axis at the beginning of the cut and H the position of the hob axis after the hob has made sixty-four revolutions. This shows that the blank must make more than one revolution in this case. If we were cutting a left-hand spiral gear with a right-hand hob, as shown in Fig. 26, the blank

would have to make less than one complete revolution for each sixty-four revolutions of the hob, the blank in this case being revolved in the opposite direction. It will thus be seen that when cutting a gear of the same "hand" as the hob, the table must revolve slightly faster than it would have to do when cutting a spur gear with the same number of teeth; but when the hob and the gear are of opposite "hand," the table must revolve more slowly than when cutting a spur gear. This has an important bearing upon the formula we are about to construct.

To gear the machine properly we must first find the ratio according to which the table is required to lag behind or lead ahead of its natural speed relative to the hob. In the first formula devised by the writer for the hobbing of spiral gears, the ratio was arrived at by considering the number of revolutions made by the hob, while the table



Machinery

Fig. 20. Diagram showing Advance Required in Table Motion when cutting a Left-hand Spiral Gear with a Left-hand Hob

makes one full revolution. The formula thus constructed for the No. 1 Farwell gear-hobbing machine is:

$$\frac{30 \times \text{No. of threads on hob}}{\text{No. of teeth} \pm [(\text{feed} \times \text{tan. of angle}) \div \text{circ. pitch}]} = \frac{P}{p}$$

This applies only to one particular machine. A later formula designed for the No. 3 Farwell machine, as shown in Fig. 24, considers the number of table revolutions required while the hob revolves a sufficient number of times to represent one revolution of the table, if we were cutting a spur gear:

$$\frac{20 \pm \dfrac{20}{\text{Pitch circumference} \div (\text{feed} \times \text{tan of angle})}}{(8 \times \text{No. of Teeth}) \div \text{No. of threads on hob}} = \frac{P}{p}$$

Being called upon to derive another formula to be used for the new No. 3 Farwell universal hobbing machine, it occurred to the writer that a formula adapted to all hobbing machines would avoid much confusion. In the following is given the process by which such a formula was derived; the result is a simpler formula than any previously used.

The "lead" of a spiral gear is the axial length of the blank in which one spiral tooth makes a complete turn around the blank. Now, in hobbing a gear with a width of face exactly equal to the lead, it is evident that the blank must gain or lose one complete revolution as compared with the number of revolutions that would be made in cutting a spur gear with the same width of face and using the same feed per revolution of the blank. Assume that we wish to cut a 30-tooth, 10-pitch, right-hand spiral gear of 45-degree angle, using a single-threaded right-hand hob and feeding 1/32 inch across the face of the blank for each revolution of the blank.

The rule for finding the lead of a spiral gear is:

$$\text{Pitch circumference} \times \text{cot. of tooth angle} = \text{lead.}$$

To get the pitch circumference, we must first find the pitch diameter; the rule for finding this in a spiral gear is:

> Pitch diameter of spur gear ÷ cos. of tooth angle = pitch diameter of spiral gear with the same number of teeth and pitch.

A 30-tooth, 10-pitch spur gear would have a pitch diameter of 3 inches. Referring to a table of trigonometrical functions we find that the cosine of 45 degrees is 0.70711; then, $3 \div 0.70711 = 4.242$ inches, which is the pitch diameter of the spiral gear. Multiplying this by 3.1416 gives 13.3267 inches, which is the pitch circumference of the spiral gear. Since the cotangent of 45 degrees is exactly 1, multiplying by this gives the same quantity (13.3267 inches) as the lead.

The next step is to find how many times the blank must revolve while the hob feeds 13.3267 inches across its face. Since the feed is 1/32 inch (0.03125 for each revolution, we can divide by 0.03125 or multiply by 32 to get the number of revolutions. This gives 426.454 revolutions. The table has been traveling faster in relation to the hob than would be the case in cutting a spur gear with the same number of teeth; in fact, the table has gained exactly one revolution on the hob. In other words, the table speed in cutting this spiral gear is to the table speed in cutting an equivalent spur gear as 426.454 is to 425.454. From this we may construct the following formula:

$$\frac{\text{Lead} \div \text{feed}}{(\text{Lead} \div \text{feed}) - 1} = \frac{\text{required table revolutions}}{\text{normal table revolutions}}$$

For a gear of opposite "hand" from that of the hob the sign would be changed to + in this formula. Use the — sign only when gear and hob are of the same "hand."

By adding a 426-tooth gear to the drivers and a 425-tooth gear to the driven gears in the regular combination used to cut a 30-tooth spur gear, we would get approximately the desired ratio, but for greater accuracy we can carry the figures to a few decimal places and factor:

$$\frac{42645}{42545} = \frac{8529}{8509} = \frac{3 \times 2843}{67 \times 127}$$

But 2843 is a prime number. We, therefore, try

$\frac{4265}{4255} = \frac{853}{851}$; but 853 is a prime number. We therefore, try

$\frac{4264}{4254} = \frac{2132}{2127} = \frac{4 \times 533}{3 \times 709}$, but 709 is a prime number. Hence we must make another slight change and try again, remembering that whatever change is made in the numerator must be exactly duplicated in the denominator to maintain the ratio as nearly as possible. The dropping of all decimals would cause a very small error, but dropping them from one side only would cause a great error. We find upon trial that

$\frac{426}{425} = \frac{2 \times 3 \times 71}{5 \times 5 \times 17}$.

Multiplying this with the change-gear combination ordinarily used to cut spur gears with 30 teeth, we have the gear combination required for any gear-hobbing machine used for cutting this gear. Thus, on the No. 3 Farwell universal hobbing machine, the spur gear ratio for cutting 30 teeth is $\frac{30}{60}$, which multiplied by $\frac{2 \times 3 \times 71}{5 \times 5 \times 17}$ gives $\frac{3 \times 71}{5 \times 5 \times 17}$, and arranging this ratio in convenient gear sizes, we have:

$$\frac{24 \times 71}{40 \times 85} = \frac{\text{Product of teeth of driving gears}}{\text{Product of teeth of driven gears}}$$

It will be noted that the last operation before factoring was to divide by the feed. Should prime numbers be encountered repeatedly in trying to factor, it is possible to get altogether new figures to work with, by making a slight change in the feed and dividing into the lead again.

Having found the gears, set the feed for exactly 1/32 inch per revolution, see that the table is revolving in the right direction, and tilt the hob spindle to bring the *thread* angle to 45 degrees and the machine is ready for business.

Recapitulation and General Remarks

The general formula for gearing any hobbing machine for generating spiral gears is thus:

$$\frac{L+F}{(L+F) \pm 1} \times \frac{P}{p} = \frac{S}{s}$$

in which

L = lead of spiral,
F = feed per revolution,
P = product of driving gears for cutting spur gears with same number of teeth,
p = product of driven gears for cutting spur gears with same number of teeth,
S = product of driving gears for cutting spiral gears,
s = product of driven gears for cutting spiral gears.

Use + sign when gear and hob are of opposite "hand," and — sign when they are of the same "hand."

In cutting teeth at large angles it is desirable to have the hob the same hand as the gear, so that the direction of the cut will come against the movement of the blank, but at ordinary angles one hob will cut both right- and left-hand gears.

The actual feed of the cutter depends upon the angle of the teeth as well as on the vertical movement of the hob. This is obtained by dividing the vertical feed by the cosine of the tooth angle; thus:

$$\frac{0.03125}{0.70711} = 0.043 \text{ inch actual feed.}$$

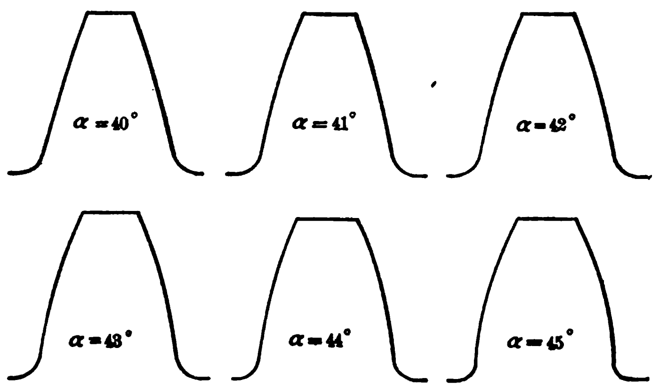
The last computation need not be made except to see that we are not figuring on too heavy a cut, as it has nothing to do with the gearing of the hobbing machine. In setting up a hobbing machine for spiral gears, care should be taken to see that the vertical feed does not trip until the machine has been stopped or the hob has fed down clear of the finished gear. Should the feed stop while the hob is still in mesh with the gear and revolving at the ratio required to generate a spiral, the hob will cut into the teeth and spoil the gear.

Should the thread angle of the hob be exactly equal to the tooth angle of the spiral gear, and both hob and gear be the same "hand," the axis of the hob spindle will be at right angles to the axis of the gear. This is in conformity with the rule that when hob and gear are of the same "hand," the hob spindle is set at the tooth angle minus the thread angle of the hob. In cutting a spiral gear to take the place of a worm-wheel, it is possible to use the same hob that was used in cutting the worm-wheel. This would be a case where it is not necessary to tilt the hob spindle. Sometimes multiple-threaded hobs are used in order to make the thread angle approximately equal to the tooth angle, when it is desired to cut spiral gears with machines on which the hob spindle swivels through only a small angle.

CHAPTER V

THE SETTING OF THE TABLE WHEN MILLING SPIRAL GEARS*

It has been frequently stated that the most suitable angle (and the one most likely to produce the best results) at which to set the table of the milling machine when milling spiral gears, is that corresponding either to the diameter of the gear measured at the bottom of the space, or to the diameter measured at the working depth. The reason invariably adduced for this is that, if the angle chosen is the angle of the spiral measured on the pitch cylinder of the gear, an undue amount of undercutting, and therefore weakening, of the teeth will



α = ANGLE OF TABLE

Fig. 30. Shapes of Teeth obtained by Setting the Table at Different Angles, Cutter and Lead remaining the same

occur, owing to an excessive amount of interference with the sides of the teeth on the part of the cutter; and that, therefore, a somewhat smaller angle should be selected to reduce these effects.

To determine whether there was, practically, anything in this idea or not, some experiments were recently made on a spiral gear, the immediate object of the experiments being to find out what the effect of altering the angle of setting of the milling machine table was upon the shape of the tooth cut.

The experiments were made upon a cast-iron gear, with a pitch diameter of 4.242 inches, and designed for 24 teeth, the diametral pitch (corresponding to the normal circular pitch) being 8. The correct cutter to use was determined by the formula $N_c = \dfrac{N}{\cos^3 \alpha}$,

*MACHINERY, Engineering Edition, June, 1911.

this cutter being No. 3 in each of the cases dealt with. The experiments consisted of cutting six teeth in the gear blank, all being of the same depth, the angle of setting of the table of the milling machine being different in each of the six cases. The spiral angle measured on the pitch cylinder was 45 degrees, the lead of the spiral being 13.32 inches, for which the gears of the spiral dividing-head were arranged. The six spirals chosen were at angles of 45, 44, 43, 42, 41, and 40 degrees, each tooth being formed by two cuts at one angle, the lead of the spiral remaining the same throughout the series of tests. It should be here noted that 43 degrees is the angle which corresponds to the diameter measured at the bottom of the space.

The profiles of the teeth taken as sections normal to the spiral on the pitch surface are indicated in Fig. 30, the profiles being drawn accurately to scale—three times full size. The various widths of the teeth at different depths were obtained as accurately as possible by means of a Brown & Sharpe gear-tooth vernier caliper. These widths are given in the accompanying table. Of course, it will be readily seen that although great care was exercised in securing measurements that would be as accurate as possible, the dimensions given above may be incorrect by about one or two thousandths inch, but not more.

TABLE OF OBSERVED TOOTH DIMENSIONS

Angle of Table Setting, Degrees	Width of Tooth at a Depth of Inches						
	0	0.050	0.100	0.125	0.150	0.200	0.250
45	0.104	0.145	0.185	0.200	0.208	0.221	0.239
44	0.102	0.144	0.181	0.195	0.202	0.220	0.238
43	0.099	0.142	0.176	0.188	0.200	0.218	0.236
42	0.094	0.135	0.168	0.180	0.196	0.215	0.234
41	0.087	0.128	0.158	0.171	0.185	0.211	0.232
40	0.078	0.115	0.146	0.158	0.171	0.205	0.230

In regard to the shapes of the teeth, it will be noticed that the 45-degree tooth is slightly undercut at the root, while the other teeth do not show any undercutting whatever. The undercutting referred to in the 45-degree tooth amounts to a reduction in width below the widest part of the tooth of about 0.010 inch.

The deductions drawn from the results of these tests are:

1. That the practice of setting the table at an angle less than the spiral or helix angle measured on the pitch surface is justified; though this angle should not be less than the spiral or helix angle measured at the bottom of the tooth.

2. That a cutter for a larger number of teeth than that given by the formula $N_c = \dfrac{N}{\cos^3 a}$ should be employed, in order to counteract the flattening and widening effect of the cutter with an angle as indicated above.

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MODERN PUNCH AND DIE CONSTRUCTION

THIRD REVISED AND ENLARGED EDITION

MACHINERY'S REFERENCE BOOK NO. 26
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NUMBER 26

MODERN PUNCH AND DIE CONSTRUCTION

THIRD REVISED AND ENLARGED EDITION

CONTENTS

CHAPTER I

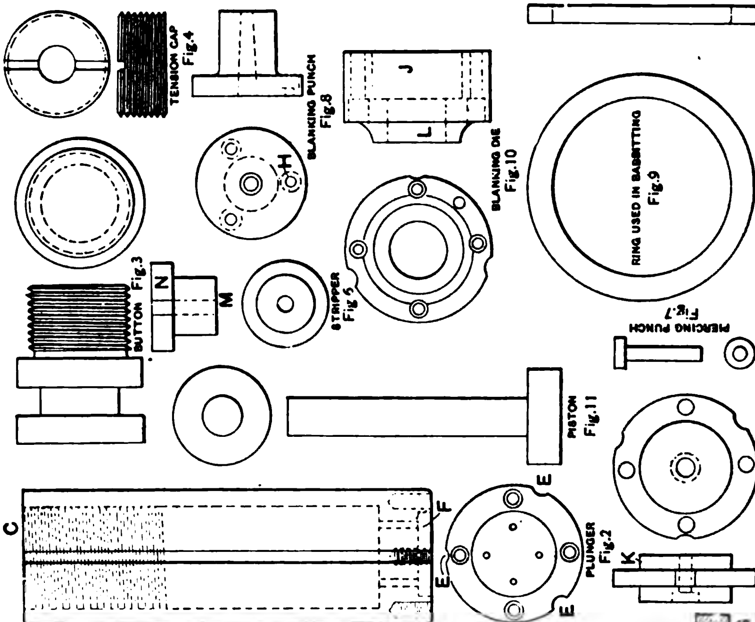
PRINCIPLES OF SUB-PRESS DIE CONSTRUCTION*

If we attempt to define the sub-press die, we find that we cannot define it as a special class of die, but merely as a principle on which all different classes of dies, cutting as well as shaping dies, may be constructed and worked. The sub-press principle is simply that the upper and lower portions of the die, the punch and die, are combined into one unit either by guide rods fastened into the lower part of the die, and extending through holes in the upper part, or by some other provision for guiding. This construction permits of a high degree of accuracy, eliminates the necessity of lining up the punch and die each time they are set upon the press, and thus saves a great deal of time and cost.

Owing to the large number of parts of which a sub-press die is composed its first cost is, of necessity, much higher than that of an ordinary die. When, however, we consider that a sub-die, when properly made, will run ten hours per day, for weeks at a time, without grinding, the first cost sinks to a minimum. In using an ordinary double die it is almost impossible to obtain two blanks that are exactly alike, one reason being that the stock to be punched is more or less wrinkled and does not lie flat on the face of the die. The consequence is, therefore, that after the piercing punches have perforated the wrinkled stock, and it is then flattened out, there is a greater distance between the holes than there is between the punches. Also, the pilot pins that are depended upon to locate the stock cannot do so exactly, since they are made a trifle smaller than the piercing punches in order to prevent them from pulling the blank up out of the die. On a certain class of work the double die answers all purposes, but when accuracy is required a sub-die is the only one that will give satisfaction.

In order to avoid a complicated drawing and to set forth the principles of the die in such a way that they may be readily understood by those not familiar with sub-dies, the die used for punching an ordinary washer has been selected for an illustration. The general principles of sub-dies are, of course, the same whether one or one hundred punches are employed. Having selected a frame with its proper cap, as shown in Fig. 1, of size suitable to the work, it is placed in a chuck, being held by the upper end, and, having faced off the bottom, the recess *A A* is bored to fit snugly the corresponding step on the base of the press. This base is finished on both top and bottom, and has a step, above referred to, turned to fit the bottom of the frame. A slot at *G* is cut in each end to receive the finger straps by means of which the frame is fastened to the face-plate of a lathe.

* MACHINERY, July, 1903.



Industrial Press, N.Y.

Figs. 2 to 11. Details of Sub-press Die

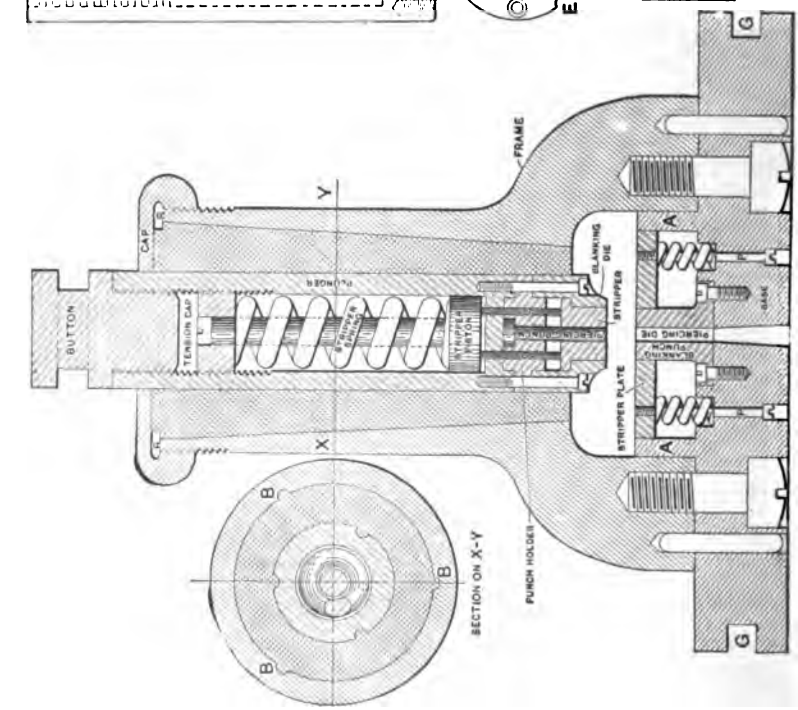


Fig. 1. Section through Sub-press Die

The center is recessed to receive the stripper plate and blanking punch, and a hole is drilled completely through to allow scrap punchings to fall to the floor. The base and frame are then fastened together by means of bolts and dowel pins as shown. Together they are clamped to the face-plate of the lathe, being centrally located by means of a plug center which fits the taper of the lathe spindle, and passes through the hole in the center of the base. In this position the frame is bored out to a taper of about one-half inch per foot. After boring, a splining tool is substituted for the boring tool, and with the lathe locked by means of the back gears, three or four grooves *B* are cut the entire length of the bore by sliding the carriage back and forth. At the same setting the upper end of the frame is faced off and threaded to receive the cap which is screwed on the frame. After the cap is in place, the hole for the plunger in this cap is bored out to the required size. This insures the hole in the cap being central with the inside of the frame.

The plunger, shown in detail in Fig. 2, is the next piece to receive consideration. After being centered and rough-turned, it is put in the center rest, and the hole *C* bored and threaded and fitted with the button shown in Fig 3. The internal thread in the plunger is carried down to a considerable depth in order to allow of the insertion of a tension cap, Fig. 4, by means of which a sufficient tension is placed upon the stripper spring to force the punching back into the stock upon the return stroke of the press. A dog is fastened to the button and the plunger turned to fit the hole in the cap, great care being exercised to keep the sides perfectly parallel. After turning, the lathe is blocked by the back gear, and three grooves *E* are splined, about 1/16 inch deep, for the entire length. It is essential that these grooves be parallel with the axis of the plunger. Before the plunger is completed, a ring, three-quarter inch wide, is made of machine steel and forced onto the lower end of it. The outside of this ring is trued up, using the plunger as an arbor, after which this end of the plunger is placed in the center rest, where the ring prevents it from being scored or injured by the center rest jaws. In this position the recess seat *F* is bored out to receive the punch holder shown in Fig. 5.

The punch holder is made, as are also the die stripper and punch, Figs. 6 and 7, by turning from a bar held in the chuck and finishing complete before cutting off. The recess which receives the head of the piercing punch should be bored at the same time to insure its being central with the rest of the die. The stripper, Fig. 6, should be made of tool steel and left large to allow for grinding after hardening, while the hole is bored sufficiently small to allow for lapping to exact size. The blanking punch, Fig. 8, which also contains the piercing die, is made of tool steel in the same manner, being finished complete before it is cut off, and it is left with sufficient stock to grind after it has been hardened. The holes *H* are drilled and counterbored for screws to hold the punch to the base.

After the parts are hardened, the blanking die is the first to be ground. It is gripped in a chuck, upper end outward, and the large

hole J is ground out to fit the step K on the punch holder. Then the hole L is ground perfectly straight and of the same diameter as the master templet. The top face is also ground off, thus completing the die. In the stripper, the hole M is lapped to the same dimension as that in the templet. A round piece of cold-rolled steel is gripped in a lathe chuck and turned to fit nicely this hole in the stripper. Without disturbing the chuck, wring the stripper onto this arbor and grind the flange or shoulder N to fit nicely the larger bore, and the smaller diameter to fit the smaller bore, of the die. The blanking punch is finished in exactly the same manner as the stripper, being ground to fit the recessed seat in the base. The minor parts, such as the stripping plate, stripper piston, pins and springs, are then made, and the press is ready for assembling.

In assembling, first force the punch holder, Fig. 5, into the seat F of the plunger, and then force the die onto the holder; transfer the holes in the die through the holder and into the plunger, and after they are drilled and tapped, fasten the parts together as shown in the sectional view, Fig. 1. Remove the die and drill four holes in the punch holder and plunger for the stripper pins O. Place the stripper piston in the plunger, above this the spring, and lastly screw the tension cap into place. The stripper pins O, which are hardened for their entire length, are placed in their holes in the punch holder, and the stripper placed in the die, which is then secured in its place on the punch holder.

The blanking punch is placed in its seat in the base and securely fastened by cap screws, after which the springs shown are placed in position and the stripper plate drawn down by means of the screws P, until it is a trifle below the top of the blanking punch. The frame is now ready to be babbitted. Screw the button onto the plunger, and with a piece of oily cloth wipe the plunger all over, then sprinkle flake graphite onto it. The oil on the plunger will cause the graphite to adhere, and after the surplus has been blown away a thin coating will be left over the entire surface. The plunger is lowered inside of the frame until the blanking punch enters the die. In the cap insert the ring shown in Fig. 9, to prevent the babbitt from flowing into the recess R, and screw the cap onto the frame. As the cap is an exact fit for the plunger, it therefore aligns it with the frame and with the blanking punch. The grooves on the plunger must be plugged with putty where they pass through the cap in order to prevent the escape of the babbitt while pouring. A pair of parallels, of a height equal to the projection of the button beyond the top of the cap, are now placed on the bench, and the die inverted upon them. Great care should be taken to avoid any vibration during pouring, as very little will affect the alignment of the plunger. Before pouring, heat the frame with a torch or jet of gas, and when the babbitt has attained the proper heat, which is a very dark red, pour it in from both sides of the die simultaneously. Allow it to remain until thoroughly cool, then remove the plunger, strap the frame to the face-plate of a lathe, and cut a spiral oil groove the entire length of the babbitt.

As the blanking punch has already been ground, the next step is to grind the faces of the blanking die, piercing punch, and stripper, while all are in their proper positions in the plunger. They should be ground so that the face of the stripper, die, and punch are all flush with each other. After grinding, the parts should be taken from the plunger and thoroughly cleaned so that no emery can possibly remain in the working parts. Oil all of the running parts in a thorough manner, then put them together in their proper positions, and replace the plunger in the frame. In setting up a sub-press die, care should be taken to have the punch come to the face of the die only, and not enter it.

CHAPTER II

CONSTRUCTION AND USE OF SUB-PRESS DIES*

The sub-press die is an old device dating back at least one, and possibly two generations, and having its origin in watch and clock factories where its ability to perform blanking operations of the most delicate nature was early recognized and fully appreciated. That this tool, though familiar in the field just mentioned, has yet capabilities in other directions which have not hitherto been fully recognized, is the impression that must be strongly borne upon an appreciative mechanic who is acquainted with the work being done in the shops of the Sloan & Chace Manufacturing Co., of Newark, N. J. This firm has for many years built precision machinery for watch makers, fine tool makers and others, whose work requires great accuracy. The tools described in the following were constructed by this firm.

A section of a typical blanking sub-press of the common cylindrical type is shown in Fig. 13. It is doubtless familiar to most toolmakers, so will need but a few words of description. To base *B* is screwed and dowelled the cylinder *A* lined with babbitt, as shown at *C*, this lining being provided with ribs which engage corresponding grooves in plunger *D* which works up and down within the babbitt lining under the action of the ram of the press in which it is used. Nut *U* furnishes an adjustment for tightening the babbitt lining to take up all slack due to wear, as fast as it is developed. The die is usually the upper member, while the punch is placed in the base. *K* is the die, screwed and dowelled to plunger *D*; accurately fitting the opening in this die is the shedder *H*, which is normally forced downward with its face flush with the face of the die by the action of spring *M*, which acts through the piston *N* and pins *O*. A similar construction is used in the bottom member. *J* is the punch, screwed and dowelled to the base. *L* is the stripper, surrounding the punch and accurately fitting it, and held firmly at the upper extremity of its movement by the pressure

* MACHINERY, December, 1906.

of the springs *Q*; it is restrained with its face flush with that of the punch by the heads on stripper screws *R*. Thus it will be seen that the faces of the punch with its stripper and the die with its shedder may be ground off smooth and flush with each other, presenting to the eye the appearance of two solid plates of metal, the division between the fixed and spring-supported members not being visible if the fitting has been well done.

With this construction in mind, the details of the punch and die shown in Fig. 14 will be readily understood. Similar letters in each

Fig. 12. Examples of Sub-press Work

case refer to similar parts, but only the members of the device actually working on the metal are here shown. The outline of the punching which is to be made will be understood from the outline of the punch and its stripper, as shown in the plan view. There are two small holes, *c c*, and one larger hole, *b*, in the blank. For punching these small holes, in addition to the simple arrangement shown in Fig. 13, openings are necessary in the punch, and small piercing punches have to be placed within the aperture of the die, passing through holes in the shedder; the holes in the punch are continued through the base of

the sub-press so that the waste material drops through beneath the machine. The piercing punches in the upper member are held to die pad *G* by holding screws *g* which draw these parts up into their tapered seats against the shoulders formed on them for the purpose. The fitting at all the cutting edges is done with great accuracy. The punch *J* fits die *K* very closely; the shredder *H* is fitted to the die very closely; the stripper *L* is fitted to the punch, and small punches *f* are accurately aligned and closely sized to their corresponding openings in the face of main punch *J*. Disappearing pins are shown at *h h*; they are used to guide the strip of stock, and are pressed down by the

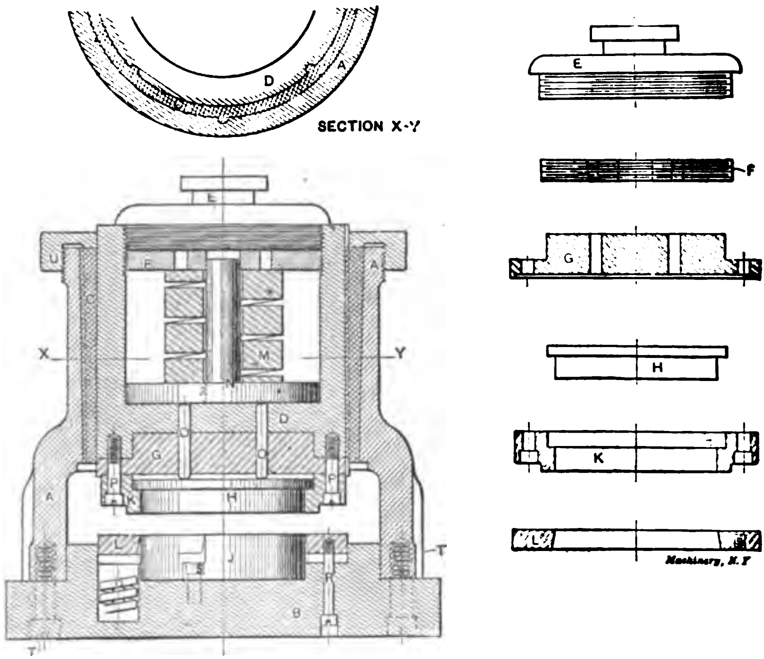


Fig. 13. Construction of Typical Sub-press

descent of die *K*, returning under the action of their springs as the ram ascends.

It will be understood, of course, that the sub-press is a complete unit, with punch and die and ram guiding surface always in place, so that no setting is necessary. The operator only needs to place the sub-press on the bed of the punch-press, insert the button on cap *E* in the holder provided for it in the face of the ram of the machine, and strap the base of the tool to the bed of the machine. He is then ready to commence work at once without any need of wasting time in matching up his dies, it only being necessary to adjust the length of the stroke to the proper amount. This is one of the advantages of the sub-press. Another one will be immediately recognized upon considering the action of the parts on the strip metal from which the

blank is punched. With the work in place, die *K*, and with it small punches *f* descend, the latter passing through the stock until they almost meet the corresponding cutting edges in the lower member. As soon as shedder *H* strikes the stock its motion is arrested, and it remains behind until the blank is separated, being meanwhile powerfully pressed upon the work by spring *M*.. As the stock, while being sheared, is pressed down around the blank, it carries with it stripper *L* which also, by the influence of springs *Q*, exerts a heavy pressure on the stock. The whole area of metal being thus firmly held between plane surfaces, there is no danger of buckling or distortion of the stock as would otherwise be likely. As the ram moves upward again the blank is still firmly held on the stationary top of punch *J* by the shedder *H*. The stock, however, is carried upward with die *K* by stripper *L*, forcing the stock back over the punching again until the movement of the stripper is arrested by the heads of screws *R*, at the time when the face of the stock is flush with the top of the punching. The work is thus pushed back into the stock in the same position that it occupied before it was severed from it and in many materials when the work has been nicely done, it is difficult at a careless glance to believe that anything has been done at all, both sides presenting a flush smooth surface where the parting occurred.

This condition is taken advantage of oftentimes in clock manufacturing. Gear blanks, for instance, are punched out from strips of metal and inserted back in their places again, minus, of course, the stock which has been punched out to form the arms and the hole for the "staff" or little shaft on which it is mounted. These strips, thus prepared, are then taken to machines where the staffs are inserted and fastened, it being much easier to handle the little wheels in this way than if they were severed and handled in bulk. A strip of stock thus treated is shown in the photograph reproduced in Fig. 12, the second one from the right at the bottom of the group; five of the pieces are shown in place in the stock while three have been pushed out. Besides the advantages of permanent setting of the punch and die and the holding of the stock to prevent distortion, which allows very narrow bridges of material to be left between wide openings, the suitability of the device for delicate work such as the piercing of small holes in thick stock will be appreciated by reference to Fig. 14. It will be noted that, no matter how small punches *e* and *f* may be, no portion of their projecting ends is at any time left unsupported laterally by shedder *H* or by the work. The shedder, pressing down firmly on the work, supports the end of the punch at the point where the pressure is applied. It is thus possible to use a very much more slender punch for a given thickness of stock than can be used in any other way. In Fig. 12, where a number of samples of sub-press work are shown, the topmost piece with the rack teeth in it, which is about 0.050 inch thick, has at its left-hand end four 0.025-inch holes pierced through it. It will be seen that the thickness of the stock in this case is twice the diameter of the hole punched. Such a ratio has perhaps been undertaken with ordinary punches and dies, although

the writer does not remember ever having seen the ratio of 1.5 to 1 exceeded; and in that case the hole in the die was considerably larger than the hole in the punch with the result that the pierced hole was very much tapered, the scrap coming out in the form of a conical plug. In the die under discussion, however, no allowance of this kind is made, the hole in the die being a very close fit to the punch, with the result that the hole pierced in the blank is as nearly a perfect

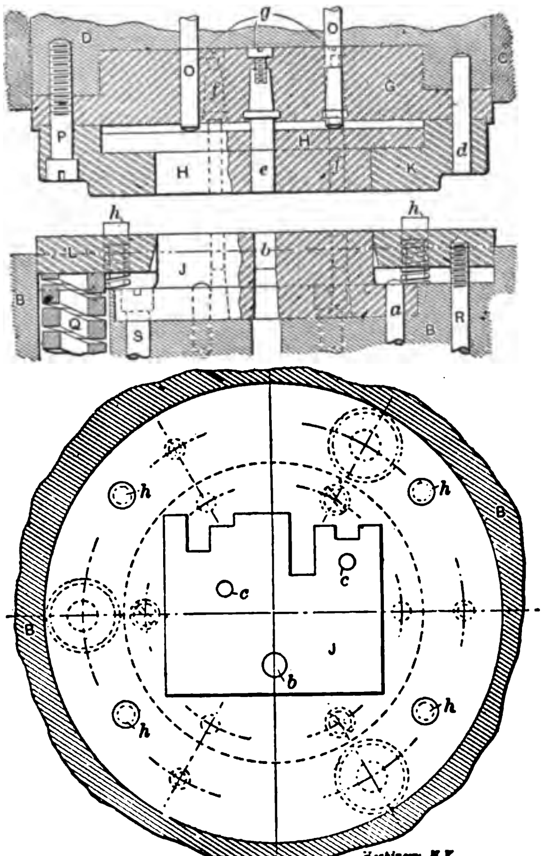


Fig. 14. Construction of Typical Sub-press Punch and Die

one as could easily be obtained by any means short of reaming or grinding.

Another advantage of the sub-press, dependent in part on the accuracy of alignment provided, and the corresponding accuracy in fitting which can be given to the cutting edges, is that the work is remarkably free from fins and burrs. A consideration of the action of the press will show that there is practically no chance for burrs to form in a piece even where they would in an ordinary blanking die.

It is, of course, necessary for the die to descend until the punch has all but entered it, if clean work is to be produced. There appears to be a slight difference in the practice of different operators in this respect, although this difference in practice would be expressed in the dimensions of only 0.002 or 0.003 inch, perhaps. Some of them adjust the stroke so that the die does not quite meet the punch. Others prefer to have them meet and even enter by an infinitesimal amount.

Attention has already been called to two of the samples of work shown in Fig. 12. The small parts there illustrated are within the ordinary range of the sub-press as ordinarily used, but it is safe to say that there are many die-makers who consider themselves familiar with this tool who have yet to see dies built on this principle large enough to blank out such a piece as the largest one shown, which is quite 14 inches square. Nor is this the limit possible. The writer saw here dies of this type being made for heavy armature work, blanking out armature segments measuring possibly as high as 26 or 28 inches across extreme dimensions. The same advantages that obtain in the smaller presses, result from the use of the larger ones. There is a saving of time in setting up the tools; there is a possibility of punching small holes in thick stock or of leaving narrow bridges of metal between openings of considerable area; the dies, owing to their accurate and permanent alignment, may be fitted to each other much more closely, produce work that requires less finishing and comes more nearly to dimensions than can be done in any other way. At the same time, the construction effects a great increase in the life of the die, making it unnecessary to grind it anywhere nearly so often as would otherwise be the case. The only disadvantage that can be set off against these advantages is the increased cost, and it appears to be conceded that even with this consideration the balance is strongly in favor of the sub-press die.

Of course the larger sizes of these tools are not made in the familiar circular form illustrated in Fig. 13. Fig. 15 shows three different styles. The one at the rear has the sliding head guided by four vertical posts carefully ground and lapped to fit cast-iron bushings. This is the construction used on heavy work. At the left is shown one in which the plunger is rectangular in shape. This works in a bearing lined with babbitt the same as the cylindrical form shown at the right of the cut and outlined in Fig. 13, although the bearing is not adjustable. The cylindrical form is used for the smaller sizes.

The making of a sub-press die requires all the skill of a first-class toolmaker. The method pursued by at least some of the men who are engaged in this work at the factory mentioned is about as follows: Taking the dies in Figs. 13 and 14 as examples, the base *B* and cylinder *A* are machined and fitted together according to methods that would naturally be pursued by any good mechanic. The inner surface of the cylinder is grooved so that the babbitt may be securely locked in place. Plunger *D* is then machined, and the outer surface ground and fluted with semi-circular grooves. Especial pains are

taken to have these grooves parallel with the axis of the plunger in both planes; if this is not done the die may be given a slight twisting movement instead of the perfectly straight forward one that is required, since upon these grooves depends the angular location of the punch and die with relation to each other. The plunger is now inserted within the cylinder and, with proper precaution, the space between them is filled with babbitt which flows into the grooves in the cylinder and those in the plunger as well, locking with one and guiding the other. After being cooled, the plunger is pumped up and down to insure a perfect bearing, and the nut *U* is screwed down until all slack is taken up. Die *K* is now made to accurately fit the templet or model furnished the tool-maker as a sample. After it has been completed, it is hardened and fastened in place. Then the model is

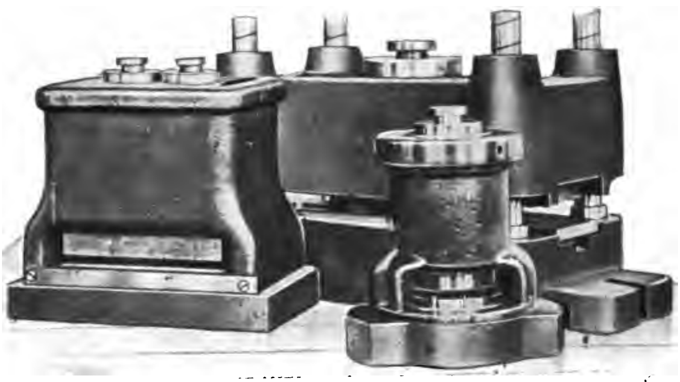


Fig 15. Three Forms of Sub-presses

inserted within it, and such holes as may be called for in the blank are transferred to die pad *G*. This is done by punches with outside diameters ground to fit the holes in the templet, and provided with sharp points concentric with the outside. The pad after being thus prick-punched, is put on the face-plate, the slight punch marks are carefully indicated, and holes are carefully bored to a taper to fit the punches which are to be inserted in them. The punches are finished by grinding on centers after they are hardened. They are supported at the shank by a male center, while the opposite end is temporarily ground to a point which revolves in a female center in the other end of the grinder. The punch may thus be ground all over with the assurance that the pointed end is true with the exterior—a necessary provision as will appear later.

It might be noted here that no draft is given to any of the cutting edges of these tools, since they do not enter each other, at least not to any appreciable extent, and since the stock in entering and leaving the cutting edges is positively moved, no clearance is necessary, and the die cuts practically the same kind of a blank at the end of its life that it did at its birth. Shedder *H* is fitted to die *K* and the holes

for the punches are transferred to it in the same way as for the die pad, by means of carefully machined prick punches which fit the holes in the models, these prick punch marks being afterward indicated to run true on the face-plate. The punch is now worked out a very slight amount larger in all its outlines than the die. The model is laid upon it, the holes transferred to it as in the case of the other parts, these holes being then indicated and bored out, but not ground in this case, being left three or four thousandths inch smaller in diameter than finished size. The punch is fastened in place in the base, lining up as nearly as possible with the die. The ram is forced downward in a screw press until the punch enters the die very slightly, cutting a thin chip from its sides to bring them to the shape required. The punch is then worked down to this point all around and again entered in the die a short distance further, the operation being repeated until the two parts fit perfectly.

In finishing the holes in the punch, after the hardening process, plugs are driven into each as shown in Fig. 16. The punches *f*, Fig. 14, still with their ends pointed concentric with their outside surfaces, are fastened in position in the upper member, and the ram is brought down until these punches mark slight centers in the top of the brass plugs, when the ram is again raised and the punch *J* removed. The punch is then strapped to the face-plate and each of the small plugs is in turn indicated from the prick punch marks, when it is removed and the hole is ground to size with a steel lap charged with diamond dust in an internal grinding fixture. The stripper is fitted to the punch in the usual manner. With the parts thus made and fitted great accuracy is obtainable.

A die of the four-posted type is detailed in Figs. 20 and 21, Fig. 21 showing the lower member or punch, while Fig. 20 shows the upper member or die. This sub-press is used in making the piece with rack teeth shown in the upper right-hand corner of Fig. 12. A slightly different method of procedure is followed in this case than with the sub-press just described. The punch and die are finished before the upper and lower members are lined up with each other. When the time comes for doing this the punch is entered in the die, the two parts being parallel with each other as to their faces, when bushings *A* are slipped over the posts until they rest in the bottom of the cast counterbores in die holder *D*, Fig. 20. This counterbored space has had pockets gouged out in the sides for the babbitt to flow into and lock with. The grooves shown in the posts in Fig. 15 are not yet cut in Fig. 21, they being still smooth and true as the grinding left them. The space *C* being poured full of babbitt and allowed to cool, the punch and die are permanently aligned with each other without possibility of shifting. The posts are then removed and the spiral grooves for oil distribution are cut in them.

One of the noticeable points about this die, as shown in Fig. 20, although the work is so closely fitted in the tool itself that the eye is scarcely able to distinguish the construction, is the fact that the section of the cutting edge which shears out the rack teeth is built

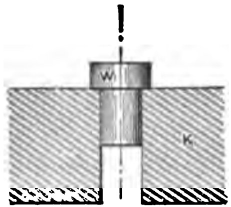


Fig. 16. Plug for Centering Holes for Grinding

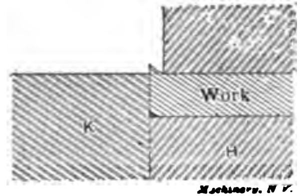


Fig. 17. Action of Badly fitting Punch and Die

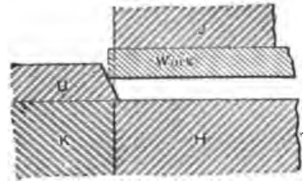


Fig. 18. "Nest" with work in Place

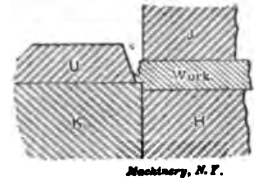


Fig. 19. Work being Trimmed in Shaving Die

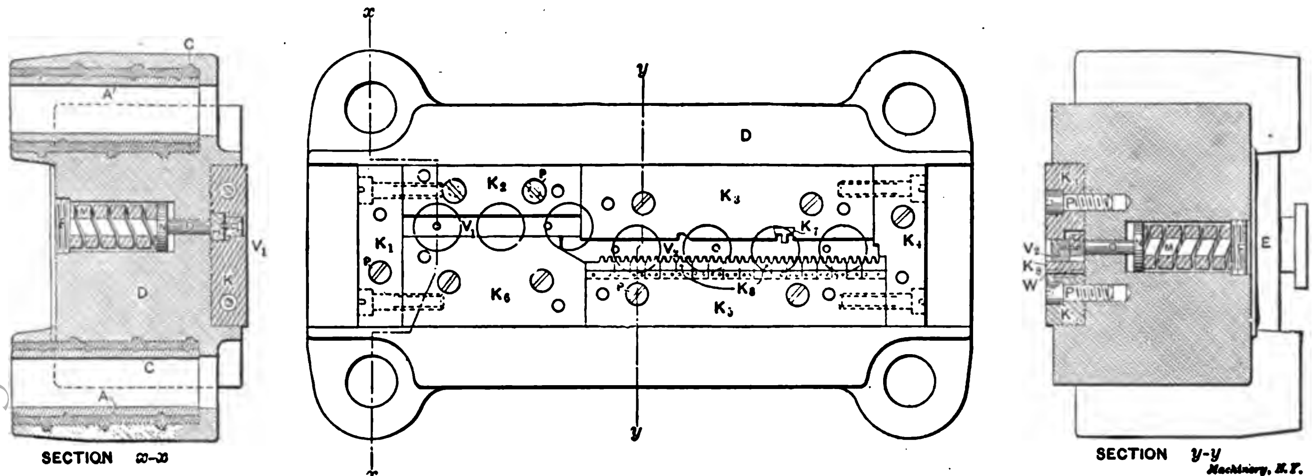


Fig. 20. Plan View and Sections of Upper Member, or Die, of Sub-press shown in Fig. 22

up of small segments, each containing two teeth only, these segments being dovetailed into the larger piece, K_6. Each of these small pieces, K_8, is secured by two dowels which pass through from side to side of K_6, locking the parts firmly together. This costly and difficult construction was necessitated by the demand for accuracy in the spacing of the teeth. With the sectional construction shown the parts are not affected sensibly in the hardening. That piece K_6 may not be warped out of shape, it is ground to size in all its surfaces, top, bottom, sides and even in the dovetail, so that when completed its plane surfaces are straight and parallel. The dovetail of the die sections K_8 are next machined to fit this and inserted, being then spaced the proper distance apart. The holes in K_6 are then continued to pieces K_8, which are taken out and hardened, and returned to be doweled in place. It will be seen that this die is constructed on the sectional plan throughout. This makes it possible to finish on the surface grinder most of the cutting edges. Troubles due to distortion in hardening are thus entirely avoided. The proper end measurements between vital points in the model are also preserved by leaving a slight amount of stock where two sections of the die come together, the parts being ground away at this point until the proper dimensions are obtained.

In the few cases where the grinding wheel will not finish the cutting surface, extended use is made of diamond laps, these being in the form of steel sections of proper contour to fit the part of the die they are working in, these steel pieces being charged with diamond dust and reciprocated vertically in filing machines, of which a large number are used in this shop. The little dovetail in which part K_7 is inserted, for instance, was finished in this way. The back of the dovetail is perpendicular but the two sides slope somewhat from the vertical, forming a wedge-shaped opening enlarged toward the rear. Section K_7 is then driven in from the rear, finished off, and ground with its front face flush with the rest of the die. In Fig. 22, which shows this sub-press, this little section has not yet been finished off, so that it is seen to project above the remaining part of the die.

This is the first operation, the tools used being the blanking punch and die. The pieces produced are afterward subjected to the action of a shaving die, the original blanks being left with 0.002 or 0.003 inch stock for the purpose, which is trimmed off in the last operation. The punch for this first or blanking die has the rack section subdivided into four parts only, which are matched up carefully with the sectional die just described. In the shaving die, however, this punch is built in sectional form as described above for the blanking die, so that great refinement in measurements is secured.

The sub-press just described is that shown at the back of Fig. 15, and opened up in Fig. 22. Its action is exactly identical with the smaller one just described; it has all its advantages and presents the same deceptive appearance of perfectly homogeneous surfaces in the punch and die when completed. In the illustration, Fig. 22, the

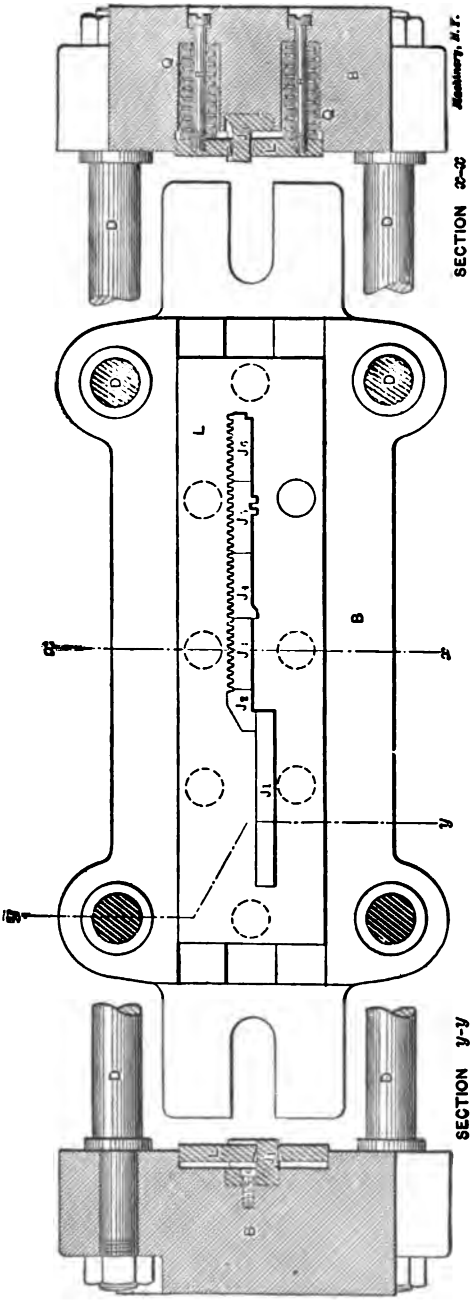


Fig. 21. Plan View and Sections of Lower Member, or Punch, of Sub-press Die shown in Fig. 22

shedder and stripper springs have been slacked up in order to show the outlines of the cutting edges, but this is not the normal condition.

A feature of the shaving die system, to which reference has been made, is the use of a "nest" to locate the work. In this trimming operation the punch is in the upper member and the die in the lower one. On the surface of the die, of which an example is shown in Fig. 23, are placed steel guiding plates, U_1 and U_2, which form the nest referred to. They have their edges shaped to the outline of the

Fig. 22. An Instructive Example of Sub-press Construction

piece to be operated upon and they are pressed inward by flat springs *W* at the outer edge, being allowed a slight lateral movement although retained from sidewise displacement by shoulder screws *V*. The holes through which these screws pass are slotted to permit this; the end of the slot limits the inward movement of the plate. As shown in the enlarged views, Figs. 18 and 19, the inner edges of these plates are beveled backward so as to form a recess in which the work may be located. The descent of the punch forces the plates out, which, as they are displaced, still guide the work so that it is properly centered over the die. These beveled edges of the plates have the further advantage of curling the chip out of the way where it does not clog the tool and may be easily cleaned off. The shedder

coming up from below and removing the work, closes the lower opening effectively so that the whole device is chip tight.

Even greater accuracy is advisable in the fitting of the punch and die in this shaving sub-press than is necessary in that used for

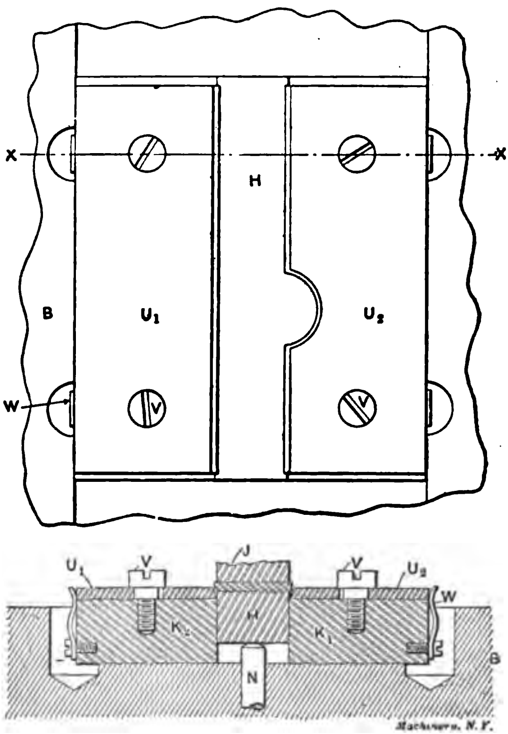


Fig. 23. Shaving Die with "Nest" for Locating the Work

blanking only, if it is desired to produce clean work free from burrs. The necessity for this will be appreciated upon examining Fig. 17, which shows in magnified form the action of the cutting edges. If the punch does not match up closely with the edge of die *K*, the stock is bent upward, leaving a sharp burr, while the punch impresses the outline of its cutting edge on the top surface of the blank.

CHAPTER III

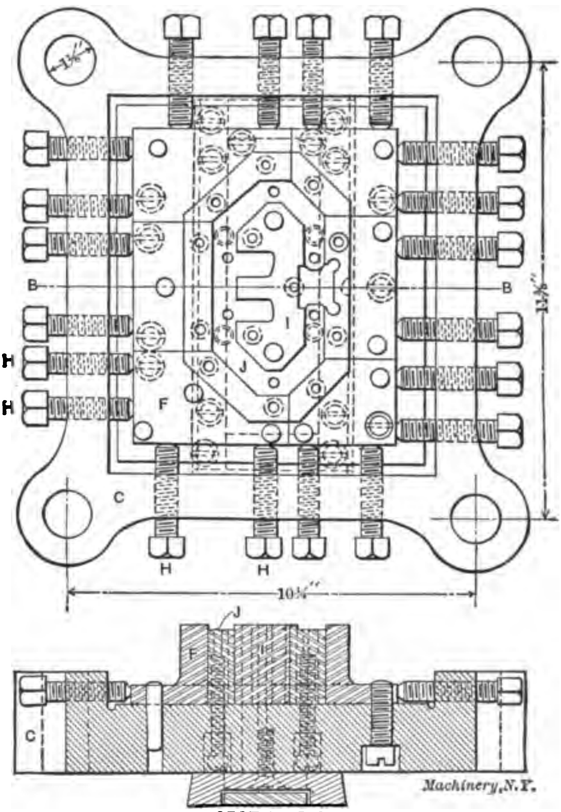
MODERN BLANKING DIE CONSTRUCTION

In the present chapter a number of designs of modern blanking dies for various purposes will be presented. It will, of course, only be possible to show general types, but the suggestions offered by the different designs will prove valuable to the mechanic who is required to design tools of this character.

Sectional Sub-press Die

In Figs. 24, 25, and 26 is shown a sectional or built-up die, working on the sub-press principle, and intended for blanking out irregular pieces, the shape of which are best indicated in the center of the punch, Fig. 25. In the assembled view, Fig. 24, the die is shown. It will be seen that the blanks can be changed to different shapes by simply inserting different die sections in different places of the die. At *A*, Fig. 26, is shown a modification of the blank, possible with this die. Another of the principal features of this sub-press sectional die is the means for stripping the scrap and ejecting, when it is wanted to produce punchings in quantities. The die shown in the cut may appear to be unduly light in construction, but several sets have been built on these lines, and have given full satisfaction. Their light weight materially lessens the cost of handling, as well as the cost of making. The holder *C* is of good, close grain cast iron planed on both sides. At the top, a recess is milled with an end mill in a vertical miller. In this recess are held the sectional parts of the die, which are fastened to the body from the bottom. After having made the necessary templets, the various die sections are shaped. A few thousandths of an inch is left on the adjoining surfaces to permit finishing by grinding. The cutting edges of the die sections must be left as hard as possible. Die section *F* is shown in detail in Fig. 26. It will be noticed that two small holes are drilled in the center of the two screw holes in the piece *F*. This is done to enable transferring the screw holes to the cast-iron holder when assembling the die. The bottoms of the die sections are left soft in order to be able to drill all the screw and pin holes through the cast-iron holder at the same setting. The dove-tailed slot in the holder *F* is made for the purpose of marking the punching. Each section is reinforced on the two outer sides by four set-screws *H*. In the center of the die a solid block *I* is fastened with three screws and two dowel pins. This block is hardened and ground all over to the shape of the templet. The ejecting or stripping device *J* for the die is made of a solid tool steel piece to the same shape as the templet, but is a very free fit, amounting to a few thousandths of an inch on the sides. This part is left

soft and is located a few thousandths inch more than the thickness of the punching below the top of the die. When the die is sharpened, the stripper is ground off the same amount. No springs are used with the stripper, it being actuated by two 1-inch studs fastened with screws on the stripper. These studs pass through the die and holder, and are actuated by a bar fastened to the gate of the press, thereby forcing out the punchings from the die. The six punches *N*, Fig. 26,



SECTION B-B

Fig. 24. Sub-press Die for Irregular Blank

are upset, as shown, at the end where they are inserted in the holder, while the other end is hardened, straightened, and lapped to size. The holes for the punches are located after the die is finished and assembled.

The cast-iron punch holder *K*, shown in Fig. 25, is planed on top and bottom and across the four bosses. The four sub-press pins *D* are of tool steel, hardened as far as the head, ground to a light driving fit on the head end, and ground to a sliding fit in the die holder on the other end. The holes for these pins were located so as to

come strictly in line with each other, and at the same time square with the punch and die. When the punch and die parts were hardened, they were placed together with two parallels placed between the castings, the punch placed inside the die, and the two clamped together with four C-clamps. In this way the holes, when bored, were bound to come in alignment.

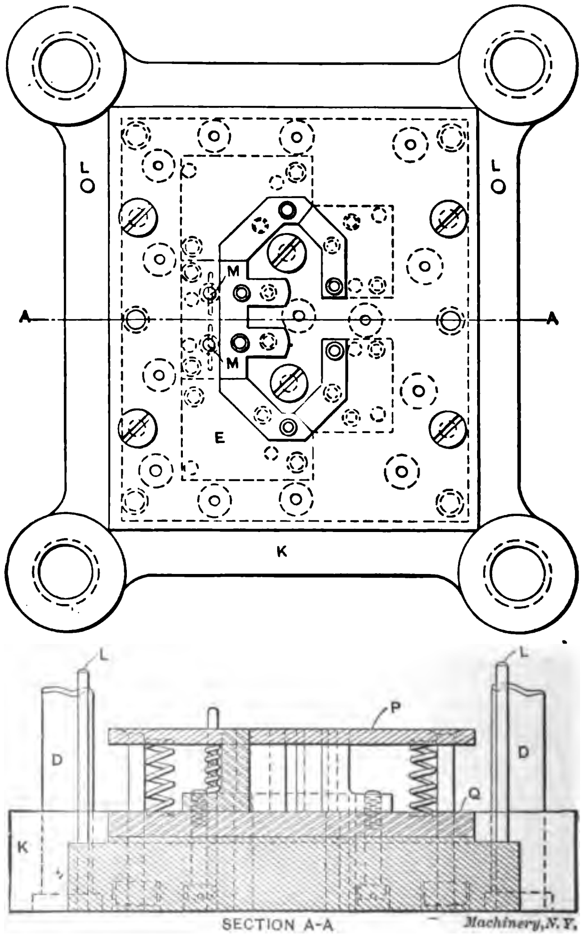


Fig. 25. Punch for Sub-press Die in Fig. 24

The punch part which is shown at *E*, Fig. 26, is made precisely as the corresponding die section, only that in locating the positions for the piercing bushings *O*, it sometimes happens that the holes for the bushings are so many and so small that they cannot be conveniently bored. The holes are then transferred by a drill that runs through the die, and is of the same size as the piercing plug, the die being used as a drill jig. After drilling, the holes are counterbored to the

right size for a driving fit for the bushings. The latter are hardened and ground all over, and the holes in them taper one-half degree. A straight pin, driven in so as to be located halfway in the bushing, and halfway in the section *E*, holds the bushing in position while in operation. A stripper plate *P* is placed over the punch sections with a free fit on both the inside and outside. It is held by flat head screws which are adjusted with nuts from the bottom of the holder. Between

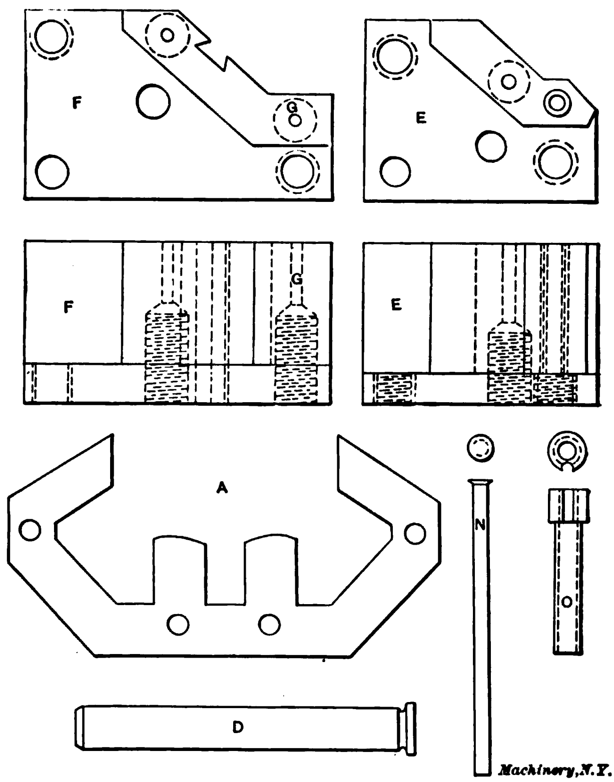


Fig. 26. Details of Sub-press Die in Figs. 24 and 25

the stripper and the punch-shoe *Q*, which is made of tool steel and hardened, sixteen spiral springs are placed to strip the metal. The punch-shoes themselves are secured with six screws to the cast-iron holder *K*.

Two guide pins *L*, Fig. 25, are driven into the top of the cast-iron holder *K*, and two gage pins *M* are located at 1/16 inch from the cutting edge. A small wire is driven through the gage pins, below the stripper, having a spiral spring underneath, which latter is seated on the punch-shoe. When the die comes down, forcing down the stripper plate, the gage pins follow, coming up again on the upward stroke.

Punch and Die for Armature Disks*

A compound punch and die for the armature disks for the cores for electric motors, having many interesting features, are shown in plan views and also in cross sections in Figs. 27 and 28. The die-holder *A*, Fig. 21, is of cast iron, and is first planed on the bottom. It is then strapped to the face-plate of a lathe, and faced and bored to re-

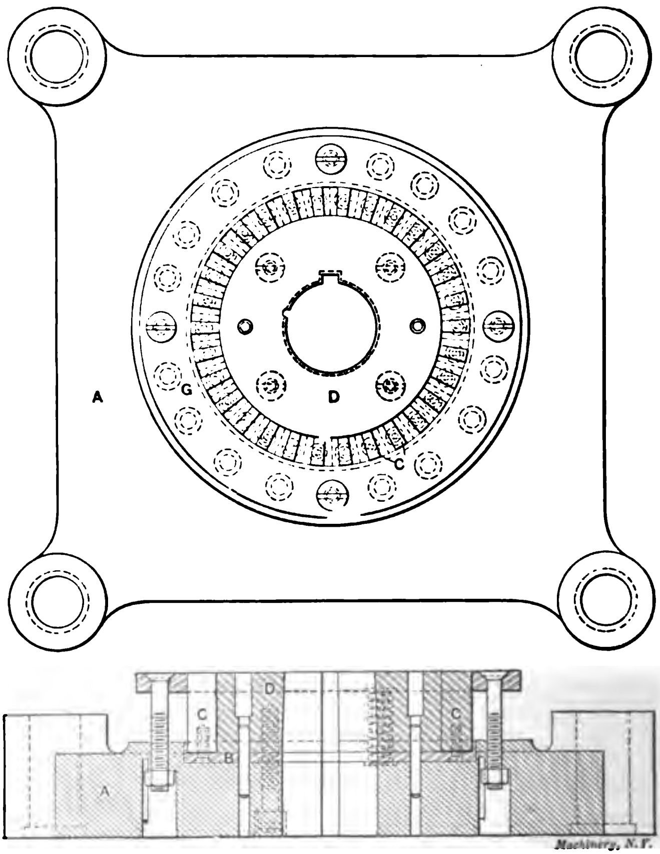


Fig. 27. Die for Making Armature Disks

ceive the plate *B*. This plate is also first faced on the bottom. It is then turned over and bored to the outside diameter of the disk to be punched; the depth of the bored hole is about ¼ inch. The die sections *C* are all milled in a fixture; they are then drilled and tapped for ¼-inch flat-headed screws. After this, the sections are hardened and tempered. The plate *B* and the sections *C* are then assembled, and after being assembled, the sections are ground. The inside ring *D* is now machined. The keyway and marking notch are

* MACHINERY, September, 1907.

tapered one-half degree for clearance; the large hole for the shaft in the armature tapers also one-half degree. The ring is drilled and tapped for four ½-inch screws, and drilled and reamed for two dowel pins. After this the ring is hardened, tempered, and ground to a very close fit in the circle formed by the sections *C*. The center hole is also ground to the required dimensions. The stripper is now made,

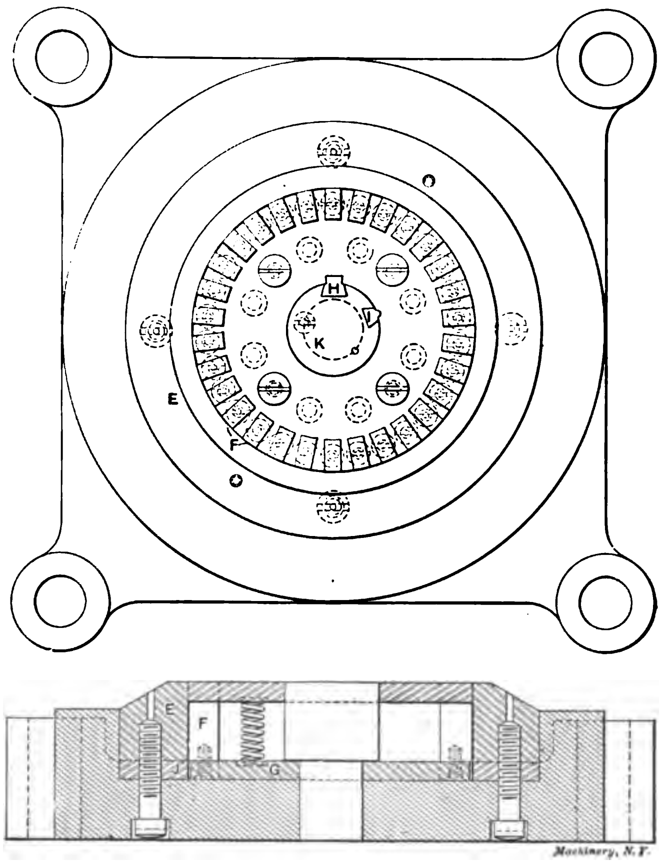


Fig. 28. Punch for Armature Disks

the working of which is plainly seen from the engraving and the whole is assembled, and the die is ready.

We are now ready to proceed with the punch. In this, *E*, Fig. 28, represents a tool steel ring, which, after it is machined, drilled and tapped, is hardened and ground. Punch sections *F* are located in the plate *G* which is milled with the proper number of slots. The punch sections should be left a little softer than the die, because the punch and die will wear much longer and give much better results

if this is the case. The sections *F* are held in place by a ring *J* which is shrunk on the outside at the bottom. At *H* and *I* are shown the punches for the keyway and the marking notch. These are fitted into the center punch, being dovetailed into this. They taper from the bottom up, when the punch is in working position, and are driven in so that when punch *K* is assembled they cannot work out. The holes for the sub-press pins are drilled and reamed with the punch and die together, and the holes in the die counterbored to a depth of

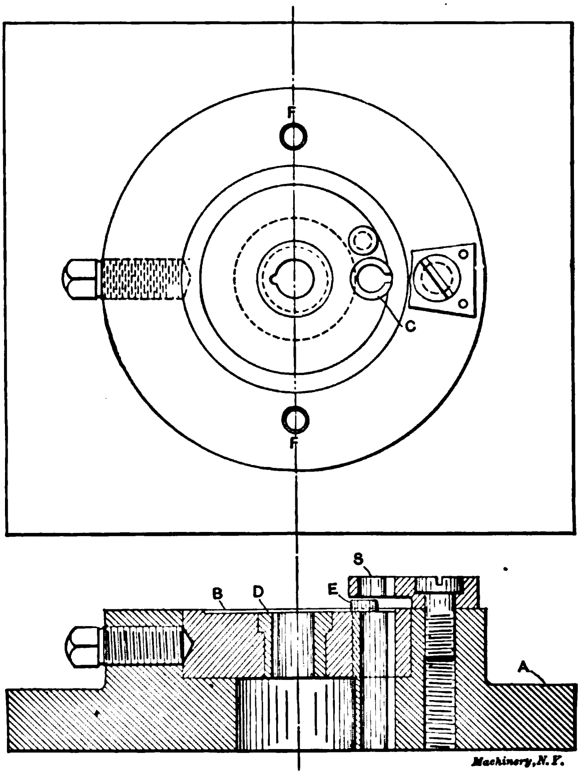


Fig. 29. Die for Armature Laminations used during Experimental Stage

about ⅜ inch. The pins should be a driving fit in the die and a working fit in the punch.

Tools for Making Armature Laminations*

The engravings, Figs. 29, 30, 31, and 32, illustrate the method of producing the armature lamination, Fig. 33, of a motor, during the experimental, and, later on, the manufacturing stage. In the first case the cost of tools is considered, and in the second, the manufacturing cost. In Fig. 29, *A* is a die holder for holding round dies.

* MACHINERY, August, 1907.

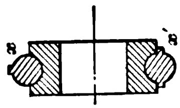


Fig. 32

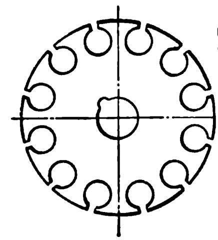


Fig. 33

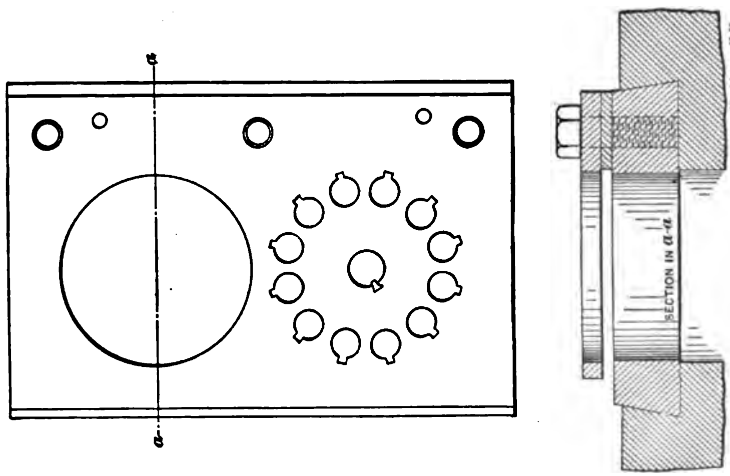


Fig. 31. Die used with Punch in Fig. 30

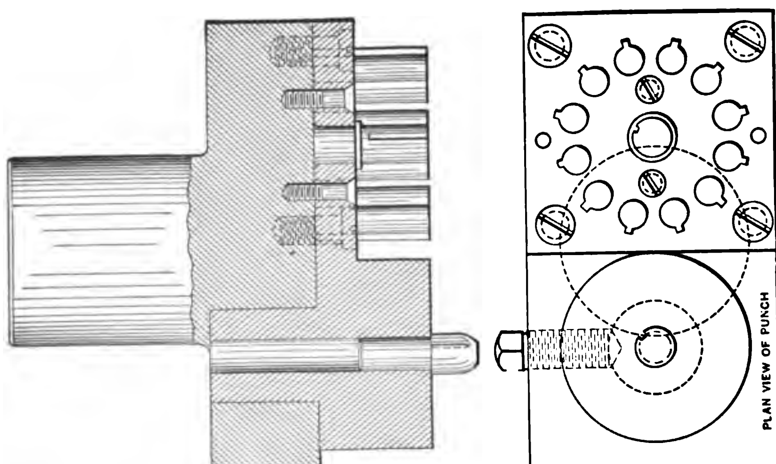


Fig. 30. Punch for Manufacturing Laminations

These holders were made for holding ordinary blanking dies, and instead of fastening the stripper to the die, it is fastened to the bolster or holder. The first operation is the punching of the blank, the second the punching of the slots. This is done with the die, Fig. 29. The pilot or index pin *E* is removed, and one slot is punched in the blank. After this operation is completed, the pin is replaced and the rest of the slots are punched, the pilot or index pin being located so as to index correctly. The die holder *B* is made from machine steel and

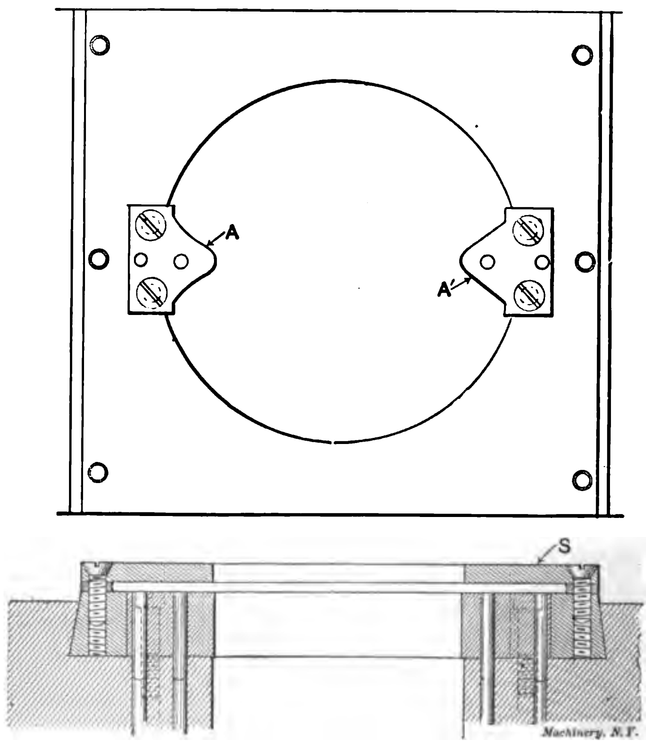


Fig. 34. Die for Outside of Blank shown in Fig. 33

recessed to allow the blank to fit properly; the die proper, *C*, is sweated fast in its place so as to avoid any chance of shifting its position. The die *D* is used for the last operation, the punching of the center hole for the shaft. The stripper *S* is removed when punching this hole, and another is fastened at *F F*. This latter is, of course, removed when punching the slots. The pilot pin *E* is also used in the last operation for locating the keyway properly in the blank. The cost of these punches and die was small, but the manufacturing cost would come high if used to produce large quantities.

As enough laminations were wanted to warrant a more expensive punch and die, and the manufacturing had to be cheapened, the de-

sign shown in Figs. 30 and 31 was adopted. These illustrations need no further explanation as to the operation of the tool. It is readily seen that a complete lamination is obtained at each stroke of the press. A special milling cutter was made to mill the punches. Fig. 32 illustrates the method of milling the punches as well as the broach for sizing the holes in the die. First both sides are milled, as shown at x', leaving a key at both sides of the punch or broach. Then one of the keys is milled off as shown at x. A small section is

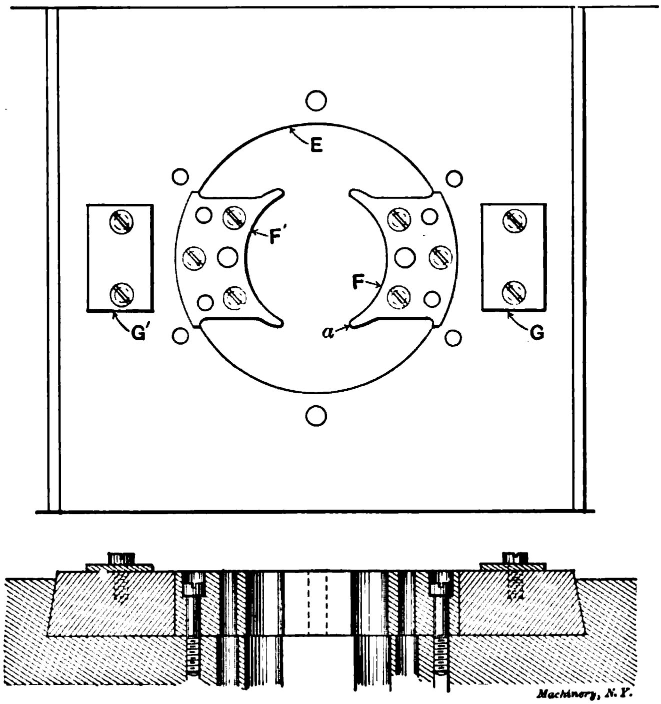


Fig. 35. Die for Inside Shape and Holes of Blank in Fig. 38

inserted at the center hole of the die, leaving a solid key in each blank instead of the keyway in the experimental lamination shown in Fig. 33.

Sectional Punches and Dies*

The punches and dies in Figs. 34, 35, 36, and 37 were made for producing the punching Fig. 38, in two operations, and illustrate to some extent sectional die making. As a perfect punching was required in regard to the inside and outside diameters, the design shown was adopted, which proved to be all that could be desired as to accuracy and cost of making, particularly when compared to previous methods and results. In making the punch and die for the first operation,

* Machinery, May, 1907.

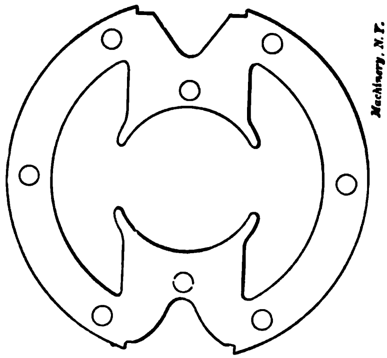


Fig. 38. Finished Punching

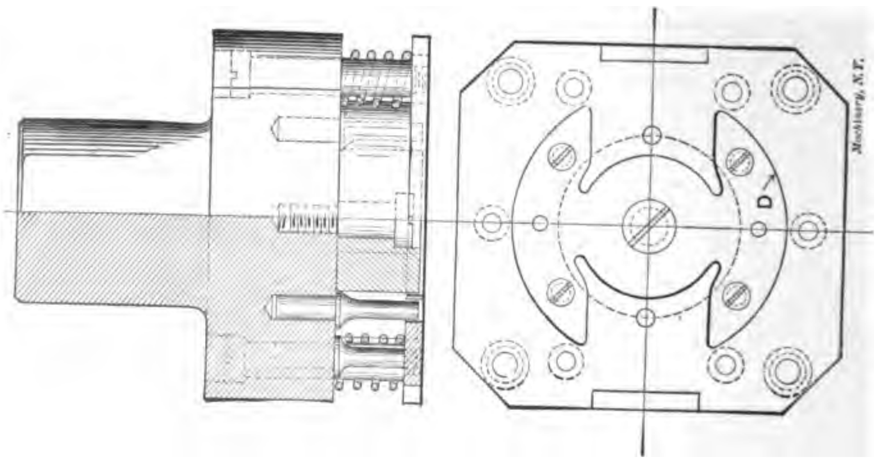


Fig. 37. Punch for Inside Shape and Holes

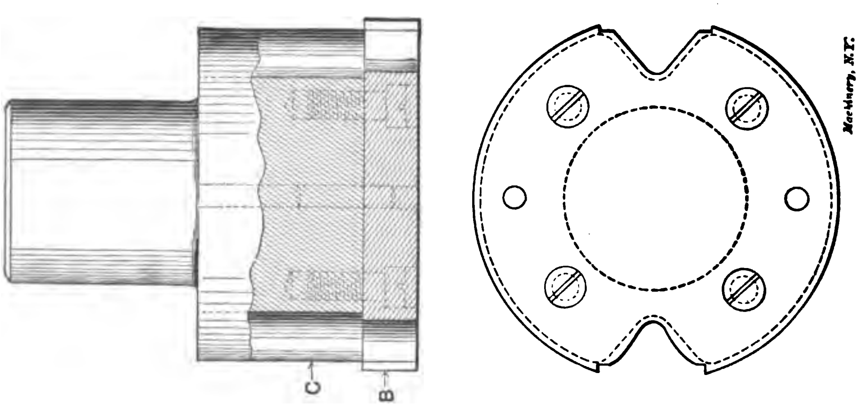


Fig. 36. Punch for Punching Outside of Blank

Figs. 34 and 36, the punch was made first. *B* is the punch proper, and *C* is the holder which is made of cast iron. The punch was hardened and screwed and doweled to the holder before grinding the outside diameter to the correct size. Then the die was machined, and after hardening ground to fit the diameter of the punch. The sections *A* and *A′* were then fitted to the die and fastened with the screws and dowel pins as shown, and sheared by the punch. As the sections *A* and *A′* were small, they did not alter any in hardening.

For the second operation, the punch and die, Figs. 35 and 37, were made in the same way, that is, the punch was hardened and ground on the diameter *D*, and the die ground at *E* to fit diameter *D*. The sections *F* and *F′* were then machined in the proper way and sheared by the punch. In hardening the sections *F* and *F′*, one of them altered so much at *a* that it had to be discarded and another made. This could have happened had the die been made solid, which would have condemned the whole die, and a new die would have had to be made. The rest of the design is readily understood by referring to engravings Fig. 34, where *S* is the stripper, and Fig. 37, where the stripper is on the punch, the blank being placed on the die guided by strips *G G′*.

Hardening Small Blanking Dies*

It is manifestly an unprofitable investment to equip a manufacturing establishment with all the latest and most approved facilities for hardening, if there are only a few pieces to be treated occasionally. In cases of this kind the diemaker must make the best of the apparatus on hand.

If a blacksmith's forge is used, let the die be placed in the fire with the cutting face upward. During the period of heating, keep the fresh coal away from the die by surrounding it on all sides and on top with red-hot cinder coal. When turning on the blast, be careful not to give it too much air. The more sparingly the blower is used, the better are the chances of the steel becoming evenly heated throughout. Turn it into the fire for about a minute, then shut it off and let the heat *soak* into the die instead of *blowing* it in. This is probably the most important point. The block of steel must be evenly saturated with heat and kept from contact with cold air until it reaches the proper hardening temperature. Remember to blow a little and then stop the air while the steel absorbs the heat. While the die is being heated, prepare a pail of clean water, taking the chill from it, that is, heating it until lukewarm. The die, held in one hand with tongs, is then plunged into the water and kept moving all the time; when the die is cool enough, take hold of it with the other hand and stir the water with it until both water and die arrive at the same degree of heat. Now instead of taking the die out of the water and reheating it over the fire or letting it cool in the air, just let the water and steel cool off together.

* H. J. Bachmann, MACHINERY, March, 1910.

CHAPTER IV

DRAWING AND FORMING DIES

Only those who have made a specialty of drawing sheet mètal know just how to proceed to lay out a die so that the desired result will be a certainty the first time it is tried. First, we are confronted with finding the diameter of the blank. There are several methods by which we can determine the size of the blank. One way is to cut out a blank of the same thickness as the stock of which the model shell is made, and then keep reducing the diameter until the blank balances the model. Another way is to multiply the circumference

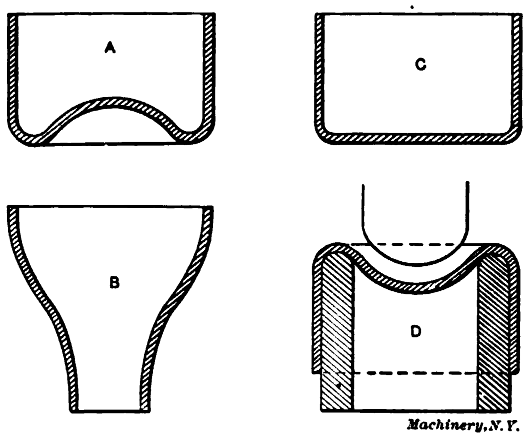
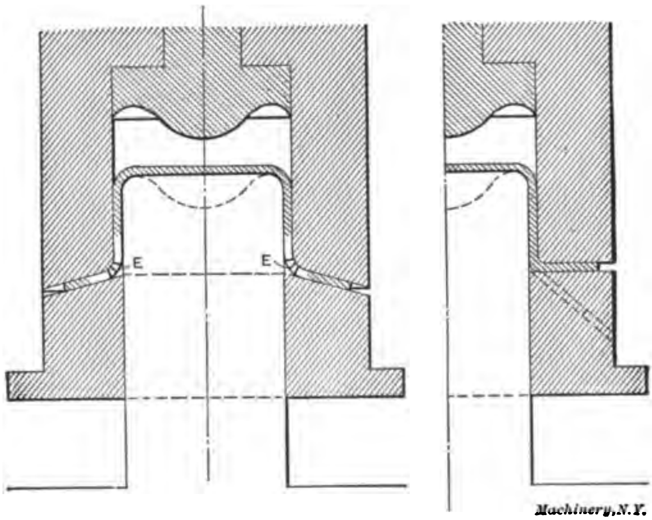


Fig. 39. Method of Drawing Shells which gives a Minimum Variation in Wall Thickness and a More Even Distribution of Stretch

by the height and add the area of the bottom, which gives the area of the blank; then find the diameter of a circle whose area is equal to the area found. This latter rule applies to blanks that must be of a uniform thickness on the sides and bottom; but if the article should be an ordinary box or something allowing a variation of thickness, the size of the blanking die can be much smaller, that is, a shell 1½ inch long can be drawn from a blank 3 inches in diameter, or from a 2-inch blank. The sides and bottom will, of course, be thinner in the latter case. A good rule to follow is to make the height of the shell at first draw, one-third the diameter of the blank, that is, a 3-inch blank, for best results, should produce a shell 1 inch long at first draw. The shells should be annealed if drawn to any length.

Fig. 39 shows a novel method when the blank is to be drawn into a shell of great length or into a shape as shown at *B*. The first operation leaves the shell as shown at *A*, and this will save at least one drawing

operation. It will be noted that the indented end in the shell bottom presents surplus stock, so to speak, whereas, if it is drawn at the first operation in the ordinary way as shown at *C*, the succeeding operations must be gentle to prevent a greater reduction in the thickness of the walls on the angular sides of the shell. Another advantage in shaping the first shell as shown at *A* is that when drawing it to the angular shape, the stretch or draw of metal is more evenly distributed over the entire surface and the strain on the metal is not as great as if the draw commenced at the corner. Another novel feature is to shape the blank as at *A* (if the shell is to be a long one) and then use a die for the second operation, as illustrated at *D*. The blank, when forced through, is actually turned inside out. This op-



Figs. 40 and 41. Diagrammatical View illustrating the Stretch of the Metal and the Effect of Angular Punch and Stripper Faces

eration presents two good points. First, we are enabled to get a suitable amount of stock in the shell so distributed as to draw to the best advantage. Second, if the shell is to be a long one, the diameter of the blank must of necessity be large, therefore the stock is distorted considerably in changing from a flat blank to a shell of much smaller diameter, and if we could see the grain of the stock as it passes over the die, it would present an appearance similar to that shown by the lines *E* in Fig. 40. That is, the inside of the stock stretches and the outside compresses. Therefore, by turning the shell inside out, the stock is apparently restored somewhat to its original texture of close grain, which will allow further stretching.

A very important point which must not be overlooked is the angle on the face of the stripper and blanking punch. In making long, small diameter shells, if the angle on the stripper and blanking punch is, say 8 degrees, the shells will break out at the corner; but by chang-

ing the angle to 10 or 12 degrees, they will come out without breaking. This point can be better understood by referring to the half section, Fig. 41. It would be almost impossible to draw the shell when the edges of the blank were at right angles to the shell, but by making the angle of the punch and stripper as shown exaggerated by the dotted lines, the shell is practically drawn by the blanking punch, and leaves very little work for the drawing punch. The above suggestions were contributed to the August, 1908, issue of MACHINERY by Mr. Frank E. Shailor.

Dies for Making Tin Nozzles*

In the following pages is described and shown a set of dies for the production of nozzles for tin cans of large sizes used to ship liquids. The dies are of the combination type used in single action presses, and perform from one to three operations at one stroke of the press. From 12,000 to 15,000 pieces of finished work can be turned out per day from these dies according to the speed of the operator.

The first die, Fig. 43, is composed of eight principal parts: *A* is a gray iron bolster plate made to separate at the line *a-b* so the die can

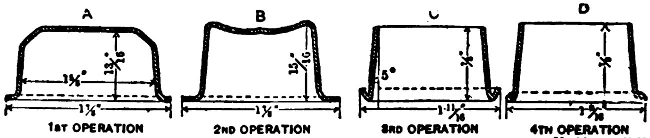


Fig. 42. Successive Operations for Making Nozzles for Tin Cans

be readily taken apart for repairs. *B* is the "cut edge" set into the top plate and held down by three flat-head screws (not shown). *C* is the center block set into the lower plate of the bolster and also held in place by flat-head screws, not shown. *D* is the pressure ring or blank-holder which rests on three pins (one shown) which in turn are supported by the washer *E*, which rests on the rubber spring surrounding the stud *F*, and held in place by another washer and nut (not shown) with which to regulate the pressure while drawing the shell. *G* is the punch and drawing die combined, the outside diameter of which is fitted to the cut-edge *B*. The inside diameter equals the center block *C* plus twice the thickness of metal. *H* is a forming pad made to fit the top of the center block *C*. It forms the top of the shell at the end of the stroke and also serves as a knock-out for the shells.

In operation the tools are set into an inclined press. The punch coming in contact with the cut-edge *B* cuts the blank, which is held by the pressure ring *D* against the end of the punch *G*, but as punch *G* continues down, the blank is drawn over the center block *C*; and, as the punch ascends, the stem *I* in the top of the punch shank comes into contact with a bar in the press pushing the pad *H* down, and the shell represented at *A*, Fig. 42, slides off back of the press.

Fig. 45 shows the second operation or redrawing tools. *A* is the bolster plate, *B* is the drawing ring, supported by pins and a rubber

* MACHINERY, May, 1905.

spring, the same as in Fig. 43. The center block in this die is tapered and the punch *F* is also bored out tapering to fit it. The pad in punch *F* is of peculiar shape, as will be noticed, and will be explained later. The shell is placed on the drawing ring, and the punch, as it descends, draws it down and compresses it to the shape of the center block *C*. The shell is knocked out on the up-stroke of the stem *H*, the same as in the first operation, and the drawn piece looks like *B*, Fig. 42.

Fig. 44 shows the tools for the third operation, which really consists of three operations. *A* is the bolster plate of the die; *B*, the trim-

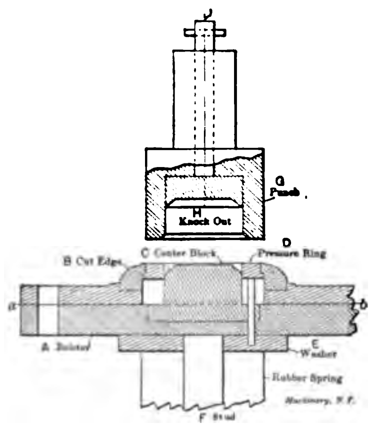


Fig. 43. First Die for Tin Can Nozzles

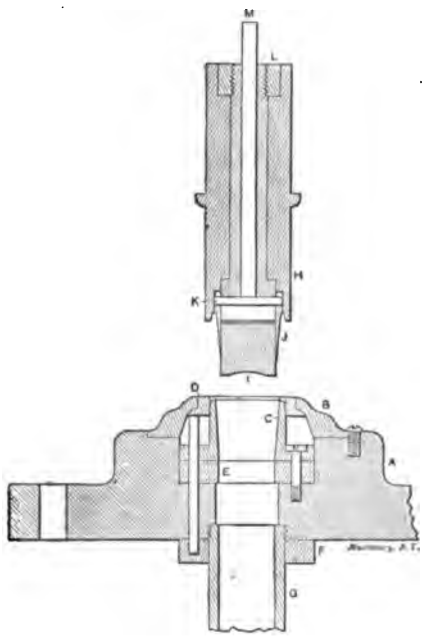


Fig. 44. Tool for Third Operation

ming die; *C*, the center block; *D*, the drawing ring; *E*, the lower die; *F*, washer; *G*, tube through which the bottom of the nozzle passes after being punched out. These bottoms are used for roofing shells for fastening tar paper in place on roofs, etc., so that in the process really two articles are made at once. These tools are used in an inclined press. As the punch comes down, punch *I* cuts out the bottom, and at the same time punch *H* trims the lap edge; as it continues downward it presses the shell over the edge of the center block *C*. As the punch ascends the knock-out bar comes in contact with the pin *M*, carrying the stripper *J* down by the cross-pin *K* and ejecting the nozzle in the shape of *C*, Fig. 42.

Fig. 46 represents the tools for the fourth and finishing operation. They consist of a simple punch and die, yet much depends on these tools,

for the nozzles all have to be of an exact size on the finished edge to receive a sealing cap, and this cap when closed on must be watertight. The die consists of a bolster-plate *A* and a die-block *B*, made of tool steel, hardened and tempered. The punch is also hardened and tempered and ground out to gage. The tools are set in the press, and the nozzle is slipped on the die-block. The punch in coming down passes over the work until the edge turned up on *C*, Fig. 42, comes in contact with the shoulder *F* on the inside of the punch. As the punch continues downward, this edge is curled over and pressed down to the

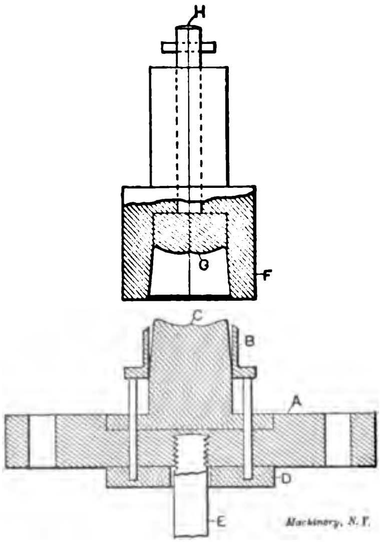


Fig. 45 Redrawing Tool for S. cond Operation

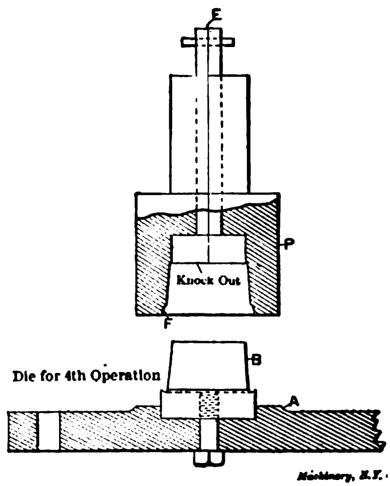


Fig. 46. Tools for the Finishing Operation

shape of *D*, Fig. 42. As the punch rises the shell is knocked out by the knock-out stem the same as in all the other dies.

Punch and Die for Blanking and Forming Copper Cups*

The die in Fig. 47 is designed to blank and form up a copper cup or capsule used in the manufacture of balance wheels for watches. The copper strip is fed into the press, which then blanks out and draws the metal into the shape shown at *R*, at the same time punching the center hole. Referring to the illustration, *A* is the base of the subpress, *B* the body, *C* the cap, and *D* the plunger, all these being of cast iron machined to size. The body and base are held together by two screws *E* after the usual well-known manner. *F* is the buffer plug which receives the thrust of the press piston, and *G* is the babbitt lining of the body *B*. *H* is the outside diameter die, held in place by four screws and two dowel pins. *H'* is the outside diameter punch, also held in place by four screws and two dowels. *I* is the die for cutting out the center hole, and *J* is the punch for this hole. *H'* and *I*

* Machinery, April, 1908.

also serve as forming dies in bringing the metal to the proper shape. *K* and *L* are shedders, supported by four push-pins, those of the former resting upon springs whose tension is controlled by short threaded plugs, as shown, and those for the latter abutting against the piston *M*, which is in turn pressed down by the large spring *N*. the tension of which is controlled by the plug *O*. The block *P* is used merely to hold the punch *J* firmly in place.

The operation of the die is as follows: The press ram being at the top stroke, the copper strip is fed in across the top of *H*, and as the

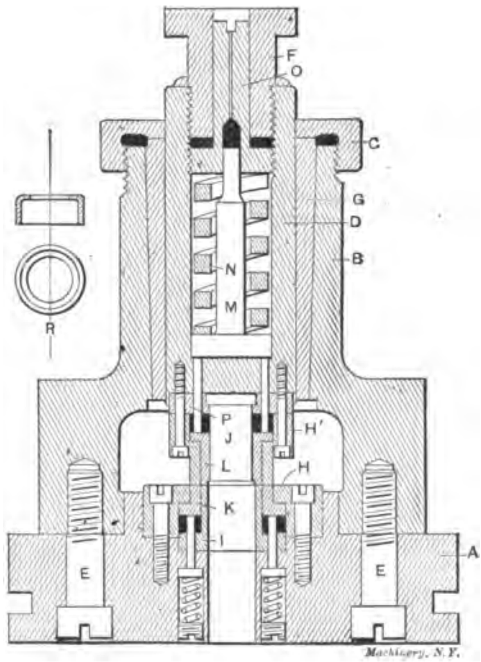


Fig. 47. Punch and Die for Blanking, Piercing, and Forming the Copper Capsule shown at R

ram descends, the blank is cut from the strip by the punch *H'* and drawn to a cup shape between the inside edge of *H'* and the outside edge of *I*. Simultaneously, the center hole is punched by *J* and *I*. As will be seen by referring to the illustration, *J* is made a trifle short so that the drawing operation will have begun before this hole is punched. This prevents any distortion of the piece by the punch *J*. Some little trouble has been experienced with this tool at first, on account of the air in the hollow plunger *D* forming a cushion when it was compressed by the rising of the piston *M*, thus preventing the proper working of the die. This was finally obviated by making a small groove at the side of the piston where it worked in the plug *O*, and drilling a vent hole through *O* as shown. This allowed free com-

munication to the atmosphere, and from then on the die gave complete satisfaction. The variation in size among the cups, or capsules, as they are called, is never more than 0.001 of an inch either in diameter or in length.

Punches and Dies for Drawing an Odd-shaped Brass Cup*

The set of punches and dies described in the following paragraphs, while not exceptionally out of the ordinary, may have one or two points that will be of interest to any one engaged in this class of work. Figs. 48 to 51 show the progressive operations from start to finish by which the piece shown in Fig. 51 is produced. This is a corrugated, conical cup, drawn from a round blank of soft brass 3/32 inch thick and 2 5/16 inches in diameter. The corrugations project externally, and internally form six equally spaced, square grooves

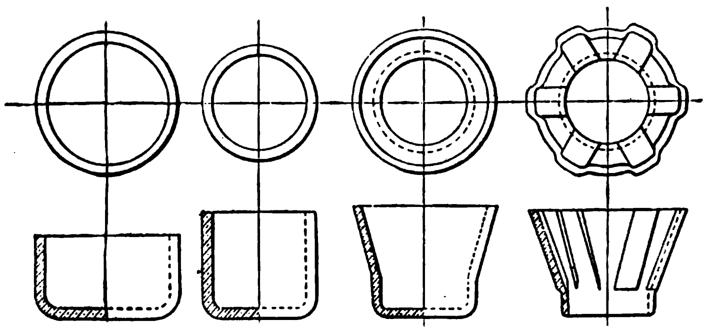


Fig. 48 Fig. 49 Fig. 50 Fig. 51

which converge radially and disappear within about 3/16 inch of the bottom. The specifications in this case require that there shall be developed on the outside a distinct shoulder at the base of the conical part and that this shoulder be formed on the corrugations only. On the inside no shoulder is to be visible, but the formed grooves are made to disappear uninterruptedly near the bottom. It will be seen that the pressure in the last drawing has to be much greater than ordinary, in order to accomplish these results, for the stock at the point where the straight and conical portions met was pressed out quite thin in order to develop the shoulder.

The dies and the shanks of all of the punches used are made of a uniform size so that two holders, one for the punches and one for the dies, are all that are required. The punches are secured by a set-screw, and the dies are seated in the holder and held fast by four set-screws, equally spaced around the side, with their points set into the circumference of the dies. To make the change from one operation to the other it is only necessary to loosen the screws, remove the tools in use, and substitute the ones next in the set. Each drawing operation is followed by a careful annealing in order to insure the equal flow of the metal, and to minimize any possibility of cracking.

* C. H. Rowe, MACHINERY, August, 1903.

The main problem is to make the finishing punch as cheaply as possible, and have it stand up under the severe work required of it. It was first made in sections by turning it to shape, and then dovetailed to receive the elevated pieces, which were made separately and forced into the dovetails up against a shoulder. Then the end was drilled and tapped, and the straight tip screwed on. The parts, having been carefully fitted, were marked, removed and hardened, and then replaced as before. The punch made in this way had been used but for a short time when the elevated pieces began to chip and crack where the strain was the greatest, and often one hundred or more cups would be run through before the defect was noticed. For this

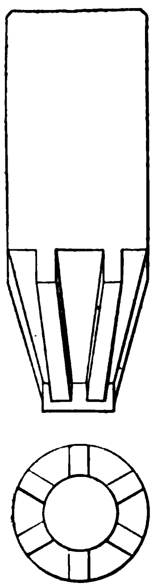


Fig. 52

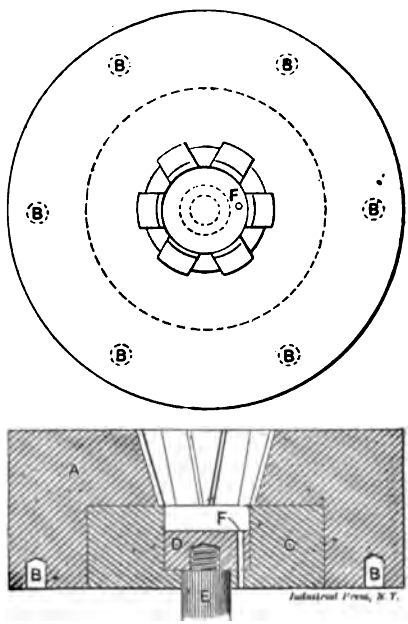


Fig. 53

reason the hardened sections were replaced by soft ones, but after a short run they would flatten down and make continual repairing necessary. To overcome the trouble the punch was finally made as shown in Fig. 52, from a solid piece of tool steel. This was milled, chipped, and filed to a finish, and after hardening was drawn to a light straw color. After this no more trouble was experienced.

The first and second drawing dies were made from tool steel disks, 5 inches in diameter and 2 inches thick, with holes to draw the cups as shown in Figs. 48 and 49. These dies were of the plain push-through type with a sharp edge on the under side to strip the work from the punch, which was made slightly tapering and of a diameter equal to the inside of the cup. In this case the punch was made less than the hole in the die an amount equal to two thicknesses of the

stock plus 0.008 inch. Previous to the last operation the punches were made considerably more than twice the thickness of the stock smaller than the die and tapered 1/8 inch per foot to assist in stripping the work, for in most cases the exact sizing of drawn work is of little importance up to the last operation. Consequently tapering the punch and making it below size will in no way interfere with the finishing of the work, which is all done in the last operation by a punch and die that must be made of suitable dimensions to meet the requirements. The rule usually adopted is to make the punch small and tapering for short cups of thick metal and nearer to size for thin stock, for the latter is more apt to develop body wrinkles if not properly pressed out in the die. The die for the third operation is made the size of the finishing die but with plain walls.

The die for producing the finished work is shown in Fig. 53. The part *A* was first turned and bored in the lathe and then fastened on a special arbor fitted to the spindle of the dividing head of a milling machine. Six 1/4-inch holes were drilled and reamed, as at *BB*, the piece being indexed so that they would be accurately spaced. A special angle iron was made for holding this piece at the angle of the sides of the die, and two holes were carefully drilled in its face to correspond with a diametrically opposite pair of the drilled holes in the back of the die. In these holes were driven 1/4-inch guide pins which projected 3/16 inch above the face of the angle iron. The angle iron was then bolted to the knee of a shaper, and the die located on its face by the pins, and strapped firm and true. The six drilled holes provided a means for accurately locating the die for planing the grooves, which was done with a formed tool screwed on the end of an extension shaper tool of the kind usually used for internal work. After shaping, the grooves were filed and polished dead smooth and the die was hardened. The piece *C* was turned and ground to a light driving fit in the back of the die and both pieces surfaced on the under side. *D* is a steel pad and *E* a stud for stripping the finished work. They are operated by the press ram on its upward stroke. The vent *F* is for the escape of air. Following the drawing of the cup the bottom was punched out in a plain cutting die.

CHAPTER V

DIE-BEDS OR BOLSTERS FOR PRESSES*

The subject of die-beds or bolsters is one of considerable importance, and is deserving of greater attention than it often receives in the shop or designing room. It has been the experience of the writer that many of the troubles encountered in the use of press tools are due to these parts being badly designed or poorly constructed. Many a fine die has been ruined because it has not been properly secured in the die-bed and consequently has shifted while in operation; or because the holes in the die-bed through which the blanks or punchings are supposed to pass have not been made large enough to allow them

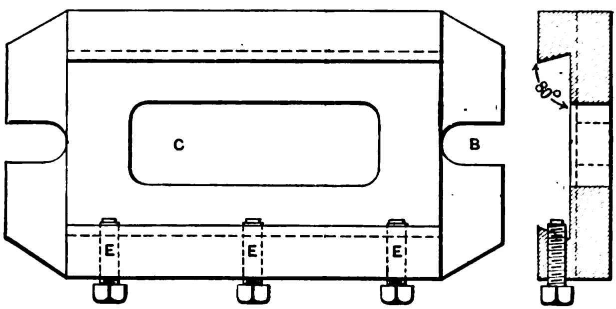


Fig. 54. Die-bed of the Style commonly used in Jobbing Shops

to pass through freely. As a consequence the blanks get jammed in the die-bed and pile up into the die itself and are compressed by the pounding of the punch, until the punch or die breaks from the strain. The principal functions of a die-bed are: first, that of supplying an adequate support for the die, and a holder to hold the die in its proper position to be engaged by the punch; and, second, to furnish a means of attachment to the press. Two of these principal points to be considered therefore in the design and construction of a die-bed are first, the method of securing the die, and second, the method of securing the die-bed to the press. Due consideration, of course, should also be given to proportion and strength.

In Fig. 54 we have an illustration of a die-bed of the type generally found in the jobbing shop. The dovetail method or holding the die, with set-screws *E* to lock it in proper position, is employed. It is fitted with a flange on each end with slots *B* to receive the clamping bolts which pass through them into the press bolster. In the center is a rectangular cored hole to let the punchings pass through. This style of die-bed is cheap and convenient for use where several dies are

* MACHINERY, January, 1910.

to be used in one die-bed. The dies can be easily slid into place and fastened by means of the set-screws, and are easily removed when another die is to be used. This bed has the following disadvantages: first, that of being hold by set-screws which have always a tendency to jar loose in punch press work, and second, the cored holes *C*, being necessarily made large to accommodate various shapes of blanks, weaken the bed and lessen the support to each of the dies. It is always better, if possible, to have a separate die-bed for each die.

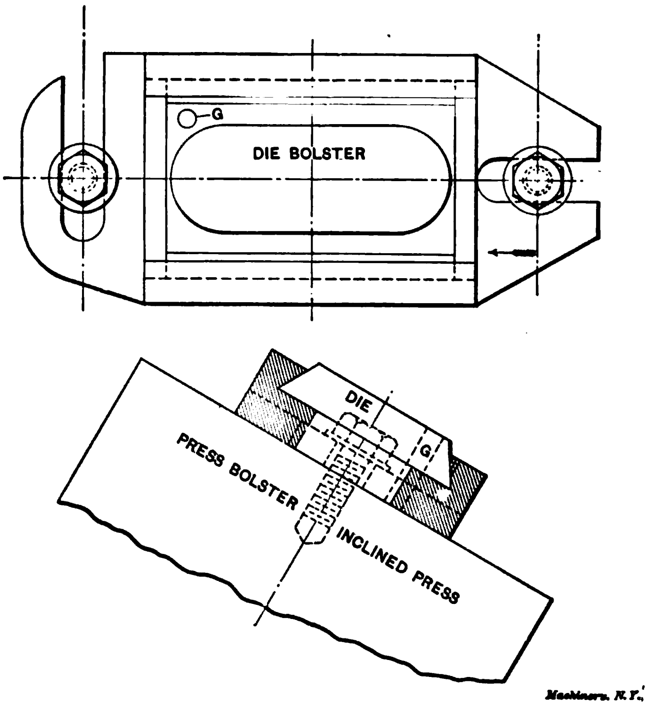


Fig. 55. Die-bed adapted to Inclined Presses

In Fig. 55 we have a bed for use on an inclined press. In this bolster the dovetail method of holding the die is used, but without the use of set-screws. The dovetailed opening to receive the die is slightly tapered and the die is driven into place with a copper mallet, and is then made doubly secure by the insertion of a dowel which is driven through the die into the die-bed. The dowel is shown at *G*. The method of clamping this bed to the press bolster is different from that shown in Fig. 54 in that the bolt slot in one flange runs at right angles to that in the opposite flange. By having the slots in this position the die-bed may be attached or removed without the necessity of taking out the bolts, thus not only saving a great deal of time and trouble in setting the tools, but also preventing the bolt holes from

getting filled with scrap or dirt and the bolts from getting lost. This is an excellent die-bed for blanking and piercing work.

An improved type of die-bed for general utility is shown in Fig. 56. In this bed the dovetail method of holding the die is used. In the illustration it will be noticed that there are four parallel pieces or gibs *E* placed along the sides of the die. The object of this is to provide for dies of various sizes. When a larger die is to be used one or more of these gibs may be taken out. This bolster, in addition to four bolt slots, has a flange *B* all around it so that it may be clamped in any position. The set-screws *H* which hold the die in place should be provided with a lock-nut as shown at *I* to lessen the chances of

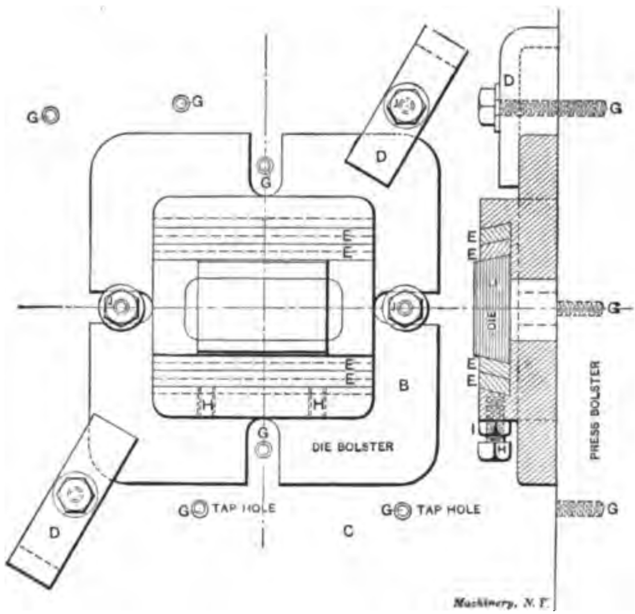


Fig. 56. Improved Form of Die-bed for General Use

jarring loose. The great advantage of having a flange all around the bolster will be apparent when it becomes necessary to swing the die-bed around enough to bring the bolt slots out of line with the tap holes in the press bolster. In a case of this kind a die-bed with a flange all around it may be clamped by means of clamps as shown at *D*, using the tap holes *G* located at different places in the press bolster *C*.

In Fig. 57 we have another die-bed of the dovetail and side setscrew variety, but with the additional feature of end-thrust set-screws. This end-thrust arrangement is an original and novel feature. In order to obtain this additional means of holding the die securely, two square grooves *B* are cut in each end of the die-bed at right angles to the opening for the die. Into these grooves a plate *C* is fitted in which

is a set-screw in such a position as to come in contact with the end of the die. With one of these plates at each end, and the set-screws screwed tightly against the ends of the die, there is less likelihood of its shifting while in operation. When short dies for simple blanking or piercing are used the end-thrust plates may be used in the inner grooves as shown in Fig. 57, and if it is desired to use a long die such

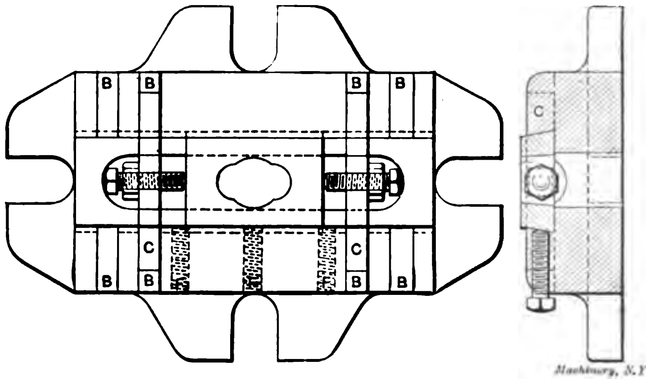


Fig. 57. Die-bed of the Dovetailed and Side Set-screw Type, also secured by End-thrust Set-screws

as is used for progressive work where there is one or more piercing operations before the work reaches the blanking punch, the plates with the set-screws may be placed in the grooves further from the center, and thus allow for the increased length of die. When the set-screws are used in these outer grooves, the heads of the screws will come directly over the slots in the flanges where the clamping bolts

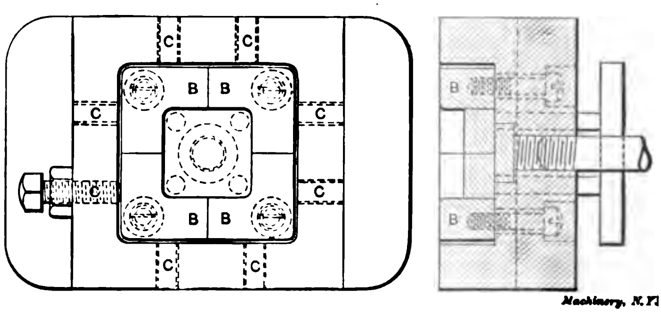


Fig. 58. Die-bed for Sectional Forming or Split Dies

should be placed; for this reason the bed should be provided with two extra slotted flanges, as shown in the illustration, to be used when necessary.

In Fig. 58 we have an illustration of a die-bed for sectional forming or blanking dies or for split dies. This bed is provided with a square receptacle to receive the dies, and with two set-screws on each side to

hold the dies in place. The square forming die shown is made in four sections *B* which are held tightly against each other by means of the set-screws *C*, and are held from working up by screws through the bottom of the die-bed—cne in each section of the die. The square recess

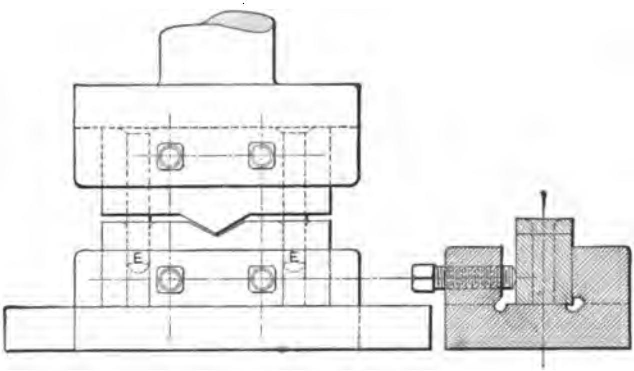


Fig. 59. Simple Form of Die Holder adapted to Bending Dies

is cast in the bed so that in preparing the bed for use it is only necessary to plane off the bottom and top of the flanges and mill the bottom of the recess, and drill and tap for the set-screws. The sides of the recess need not be machined as the dies have no bearing on them.

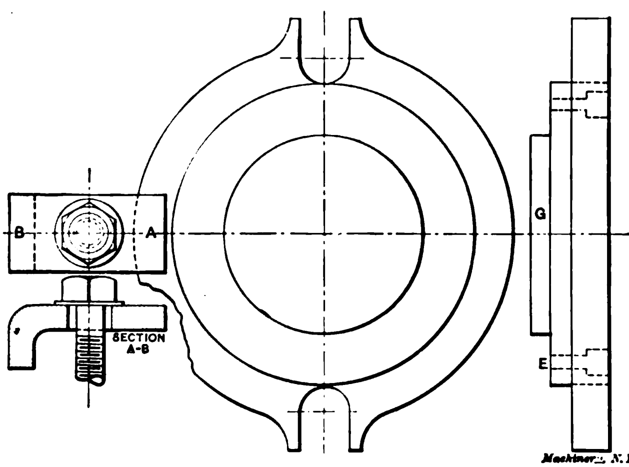


Fig 60. Die-bed or Bolster for Round Drawn Work

A very simple type of die-bed for bending and forming dies is shown in Fig. 59. It is simply a vise similar in some respects to a milling vise, but having two set-screws to take the place of the movable jaw. The die is simply set in the bed and clamped against the solid jaw by

means of the set-screws. This type of bolster is intended for use only on dies that do not require a "push up," and when the bending or forming operations are done on a solid surface. In order to obtain the best results from this die-bed, the complete outfit of punch holder, punch and die of the type shown in the sketch should be used. The punch holder and punch are made just the same as the die and die-bed. They are kept in alignment when in operation by the two guide pins *E* which are secured in the punch and which enter the die at every stroke of the press, making it practically impossible for the tools to shift while in operation. If it be desired to change the tools it is not necessary to disturb the punch holder or die-bed. They may be left in the press, and by simply loosening the set-screws in the die-bed and punch holder, the punch and die held together by the guide pins may be taken out and set aside and another set slipped into their places.

Fig. 60 represents a bolster for combination dies for round drawing work. This bolster requires but little explanation. It is circular in shape with two steps or extensions, two bolt slots and a flange all around it to allow it to be clamped at any convenient place. When the combination dies are turned in the lathe the bottom die is counterbored to be a driving fit on the extension *G*, and is held down by screws that pass through the bed at *E* into the die.

CHAPTER VI

FEED STOPS FOR PRESSES*

The simple feed-stops here illustrated are not new or novel in their construction; experienced toolmakers will recognize them at once as "old acquaintances," but there are certain points concerning them of which an explanation will be of benefit to those who are not experienced in punch and die work.

Fig. 61 shows a fixed stop-pin *C*, which is the most common of all feed-stops. It is the particular form of this simple stop to which attention is called. The common way to make a fixed stop-pin is to bend over a piece of steel rod and drive it into the die. This appears simple enough, but it is not so simple as it looks. The difficulties and disadvantages connected with making a bent stop-pin are as follows: First, the difficulty of bending the pin at right angles without breaking it or bending the part to be driven into the die; second, after the pin has been made and hardened it is apt to break in driving it home to its place in the die because of its uneven shape; third, in driving the pin into the die it is apt to swing around out of its proper position making it necessary to knock it around again and thus increasing the chances

* MACHINERY, September, 1909.

of breaking it. Every time the die is ground, this difficulty is experienced and the result is frequent breakage and consequent loss of time in waiting for new stops. All these difficulties are overcome by making the style of pin shown in Fig. 61. This is simply a shoulder pin turned to a nice snug fit in the die. The shoulder, which acts as the stop for the stock, may be made larger or smaller in diameter according to the width of scrap desired between blanks. This stop is quickly and easily made, is easily taken out and put back again after grinding the die, and it will last as long as the die itself. It is a good idea to cut a hole through the stripper *A* directly over the stop-pin as shown at *G* so that the operator can see the pin when the press is in operation.

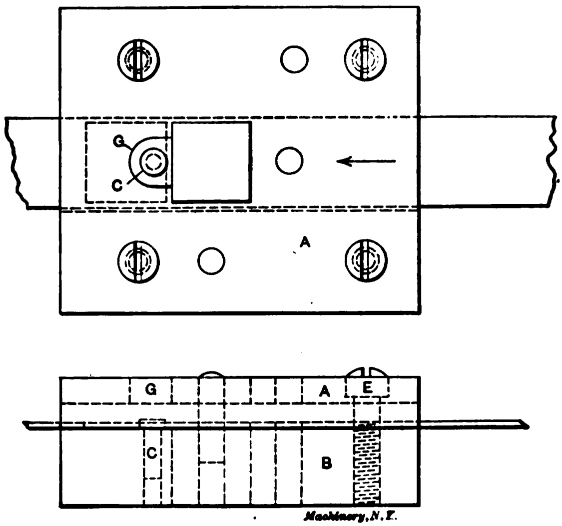


Fig. 61. Latch Stop-pin of Simple Design

The stop shown in Fig. 62 is excellent because of its simplicity, and also because of the great variety of work to which it may be applied. This stop is of the latch variety, but it differs from most stops of this type in that it requires no mechanism to lift it. It is not operated by the action of the press nor by the punch, as is generally the case with latches. Its construction is simple. A hole is drilled through the stripper *A* to receive the pin *K* which passes through a hole in the stop *C*. The stop swings upon this pin. A light flat spring *D* is fastened to the top of the stripper so that the end of the spring rests on top of the stop. In securing this spring to the stripper, it is only necessary to place one end under the head of the screw *E* with a piece of the same material under the opposite side of the screw as shown in the plan view. By this method the spring can be quickly and easily attached or removed, and a straight piece of spring material can be used. The stripper, of course, should have dowel pins in it to insure its coming back into the same position every time the die is ground. The dowels

and screws are shown at *H* and *G*, respectively. The stripper should also be cut off at the stop end as shown at *L* so that the stop will be outside of the stripper and in full view of the operator. The action of this stop is as follows: The stock *F* is fed to the left, and as the punched strip passes the stop, the point of the stop *M* drops or rather springs into the hole made by the blanking punch. The operator then pulls the strip back against the straight outer edge of the stop, and holds it there until the next blank is punched. This process is re-

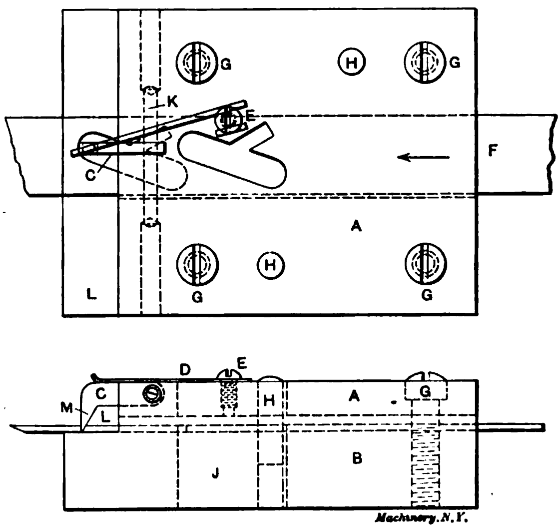


Fig. 62. Improved Form of Fixed Stop-pin

peated at each stroke of the press, the scrap between the blanks being pushed past the stop each time and then pulled back against it. The inner beveled edge of the point *M* causes the stop to lift as the scrap between the blanks is pushed against it, while the outer edge, which is at right angles with the die, prevents the stop from lifting when the edge of the scrap is pulled back against it.

By this simple stop the operator can feed the stock at will without waiting for the operation of a mechanically-lifted stop, to say nothing of the time that is saved by not having to adjust an automatic stop. An operator can make about 40,000 blanks per day with dies fitted with this form of stop on a press making about 100 strokes per minute. These stops are used only on hand-fed work.

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By GEO. L. FOWLER and CARL J. MELLIN

PART I.
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No. 27—LOCOMOTIVE DESIGN

By Geo. L. Fowler and Carl J. Mellin

PART I.

BOILER AND CYLINDERS

CONTENTS

CHAPTER I.

PRELIMINARY CONSIDERATIONS.*

Of all the various departments constituting the working elements of a railroad system, that of the motive power is undoubtedly one of the most important, and it is becoming to be more and more recognized that the greatest care and the highest skill must be exercised in order to obtain that degree of efficiency that the modern management demands. That this may be done necessitates not only extensive experience and an intimate knowledge of the general principles of mechanics and engineering, but a sound judgment as to the peculiar characteristics of the locomotives and rolling stock that shall adapt them to the conditions of the road over which they are to be operated.

It is, therefore, the duty of this department to thoroughly investigate the character of the freight to be moved, the strength of the bridges as well as the peculiarities of the roadway, as a basis for forming an intelligent opinion as to the power to be provided to comply not only with the present conditions imposed by the operating department, but to allow for future expansion and development.

The object, then, of this treatise is to prepare a simple and comprehensive guide, supplemented by examples from practice, by which the type and size of an engine for a definite service may be determined. In order to convey some appreciation of what it means to design a locomotive that will be well adapted to economically perform the work which it is desired that it shall do, a few words will be of value, to set forth some of the many and varying influences that have a bearing on the problem. The officers in charge of a railroad may have preconceived ideas as to the economical value of the heaviest types of engines for the hauling of a high tonnage, but they will almost invariably find themselves limited in the matter of the application of such engines by the weight of the rail that is down, the strength of the bridges, and the clearances of the permanent way.

Added to these, the grades, the quality of the coal that will be available, the speeds and character of the traffic are modifying factors that will affect not only the final details but the type of the design. It must be understood also that, despite the scrutiny and study to which the locomotive has been subjected, there is still much to be learned and a vast amount of research must be brought to fruition before the designer will have all of the data that he needs for the task to which

*The present number of MACHINERY'S Reference Series is the first part of a treatise on complete Locomotive Design, covered by Nos. 27, 28, 29, and 30 of the Series, and originally published in RAILWAY MACHINERY (the railway edition of MACHINERY). Each of the four parts of the complete work treats separately on one, or more, special features of locomotive design; and while the four parts make one homogeneous treatise on the whole subject, each part is complete by itself.

he has set himself. It is, therefore, in view of these limitations that it is impossible to formulate any set of hard and fast rules that can be made to serve and which will be accepted as absolutely correct in all quarters.

In order, then, that what follows may serve somewhat as a guide to show what has been done in certain concrete cases, a number of assumptions will be made as a basis of work, and two locomotives will be worked out and developed that will meet the requirements of this supposititious road and the assumptions that will be made in connection with its operation. It will be understood that any variation from the assumed conditions may cause modifications of design, that will be more or less extensive according to the character of the variations.

In order that a start may be made, we will suppose that two locomotives designed respectively for freight and passenger service are to be designed for a division one hundred and fifty miles in length, that is laid with rails weighing 75 pounds to the yard, whose bridges are of such strength that they are up to the full capacity of the rail,

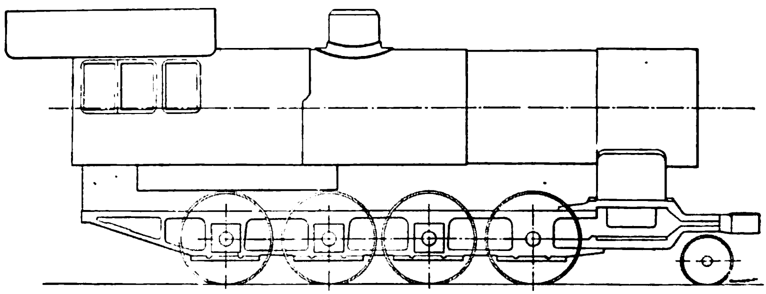


Fig. 1. Outline of a Consolidation Freight Locomotive.

with clearances that permit the usual widths and heights of design, and finally with a ruling grade of one per cent 10 miles in length. These figures, scanty as they are, will suffice as a guide to indicate to the locomotive designer the conditions that he must meet. It is, of course, beyond the province of this work to enter into a discussion of the construction of the track, so that it will be merely taken for granted that this is of the most approved character and is kept in first-class condition.

With this data at hand, the requirements usually sent to the locomotive designer are that he shall supply an engine that will haul a given tonnage over a ruling grade at a minimum speed. In the drawing up of the specifications in this form, judgment, backed by experience, must be exercised that the requirements do not call for a heavier engine than the rails are able to carry.

The first steps, then, in the determination of the size and character of the engine that can be used is to ascertain the weight per wheel that can be safely carried upon the rail. This depends upon the metal of which it is formed, the shape, and size or weight. If we take the fiber stress to be put on the rail under a static stress as 12,500

pounds, we find that a wheel load of 22,000 pounds will meet the requirements for a 75-pound rail, and this experience has shown to be good current practice.

For heavy freight work on roads of ordinary curvature, it has been found that four driving wheels coupled are about as many as can be satisfactorily worked. Engines with larger numbers have been built, but even those roads using them have reverted to the consolidation type, having four pairs of wheels coupled and a pony truck in front.

In the case of our supposititious division then, the type that current practice would suggest for adoption would be a consolidation engine, and the weight upon each driving wheel according to the assumed conditions would be about 19,400 pounds, with 21,000 pounds on the truck, or about 176,000 pounds for the total weight of the whole machine.

Experience has shown that the tractive power of an engine can be made from 22 to 26 per cent of the weight on the drivers. In the present case we will use the former as suitable to the designs to be

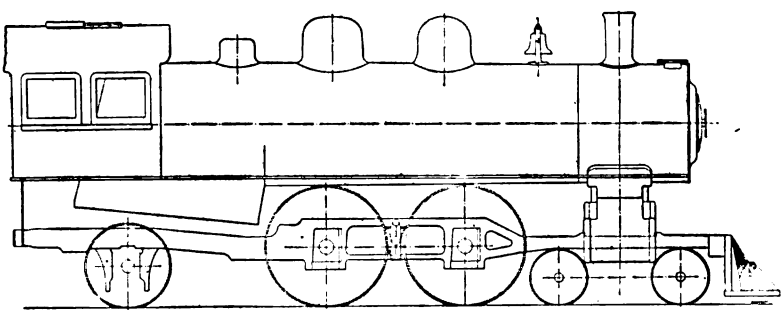


Fig. 2. Outline of an Atlantic Type Passenger Locomotive.

presented, which would make a total tractive force of 34,200 pounds from which 10 per cent should be deducted for the internal friction and resistance of the engine, leaving 30,800 pounds or about 31,000 pounds available tractive power. When an attempt is made to refer this drawbar pull to the weight of train that can be hauled, we find at once a mass of variable resistances that again render accurate calculations an impossibility. Train resistances will vary with the number of cars, the conditions of the journal lubrication, the direction and force of the wind, and many other minor details of the construction. The best that can be done then is to take an approximate formula for train resistance and make due allowance for the excess resistances resulting from emergencies.

There are some variations in the formulas given by different authorities. The work of foreign specialists is of little or no value to American designers on account of the difference in the rolling stock experimented with, and of all the work done in this country the formula known as that of the *Engineering News* is the most widely accepted for approximate reliability. This formula is

$$R = \tfrac{1}{4}\, v + 2 \qquad (1)$$

in which R is the resistance in pounds per ton of 2,000 pounds in the weight of the train, and v the speed of the train in miles per hour. By substituting a value for v of 10 miles per hour, in the example under consideration, we find the rolling resistance to be

$$R = \frac{10}{4} + 2 = 4.5 \text{ pounds per ton of 2,000 pounds.}$$

The resistance due to grade is found by the formula

$$R' = l \times \frac{p}{100}, \text{ in which} \tag{2}$$

R' = resistance in pounds,
l = load in pounds to be carried up the grade,
p = percentage of gradient.

From this it appears that the resistance per ton of load on a 1 per cent grade is

$$R' = 2{,}000 \times \frac{1}{100} = 20 \text{ pounds.}$$

To these resistances should be added the resistance due to curves, which is found to be about 0.5 pound per degree of curvature per ton of load where the curves are not compensated for in the reduction of the grades. On modern roads, however, the curves are generally compensated for in the gradient and it will therefore be omitted in the example and the rolling and grade resistances only used. Hence $R + R' = 4.5 + 20 = 24.5$ pounds to the ton, leaving the minor uncontrollable resistances to be taken care of by a certain margin provided for such purposes and other emergencies.

We have seen that the available tractive power of an engine to do the required work will have to be 30,800 pounds under the driving wheels, but it would not be advisable to load the engine to these figures under all conditions without allowing a reasonable margin for the uncontrollable resistances, previously referred to, in the variation of car resistances and weather conditions of about eleven per cent, which leaves 27,400 pounds for moving the train, inclusive of the weight of engine and tender.

By dividing this amount by the resistances obtained from formulas (1) and (2) we get the total weight of the train to be hauled up the one per cent grade at the rate of ten miles per hour, *viz.:*

$$\frac{27{,}400}{24.5} = 1{,}118 \text{ tons of 2,000 pounds.}$$

Before it is possible to determine the load behind the tender, it is necessary to work out the dimensions of the engine and thus ascertain the weight of the engine and tender. With the requirements before us experience has taught that the following dimensions of wheels and their related parts will most satisfactorily meet the conditions, namely:

Diameter of driving wheels, 57 inches.

Boiler pressure, 200 pounds.

Stroke of pistons, 26 inches.

With these dimensions given, we get the cylinder diameter from the formula:

$$d = \sqrt{\frac{T \times D}{P \times 0.85 \times S}}, \text{ in which} \qquad (3)$$

d = diameter of cylinder,
T = the required tractive power,
D = diameter of driving wheels,
P = boiler pressure,
S = stroke of pistons.

As 85 per cent is the generally adopted coefficient of the boiler pressure for average cylinder pressure at low speed, that is taken as a constant factor. By substituting the values decided upon in the formula we obtain the cylinder diameter

$$d = \sqrt{\frac{34{,}200 \times 57}{200 \times 0.85 \times 26}} = 21 \text{ inches.}$$

The weight allowed on the driving wheels is 155,000 pounds and by ordinary proportions in the design of an engine of this type about 21,000 pounds will come on the truck, making a total weight of the engine alone of 176,000 pounds, and the weight of the tender filled with coal and water will be about 110,000 pounds, which added together, makes 286,000 pounds or 143 tons.

This amount will now be deducted from the previously obtained total weight of the train to find the load behind the tender, namely: 1,118 — 143 = 975 tons.

A rational specification for an engine to work on such a grade, then, would be one capable of hauling a train of 975 tons at a speed of ten miles an hour, leaving a reasonable margin to be utilized under favorable conditions. The result will be a consolidation locomotive of the general outline shown in Fig. 1.

Turning now to the matter of the passenger locomotive, there are three types in common use in this country. They are the eight-wheeled American or 4-4-0 type, the Atlantic or 4-4-2 type, and the ten-wheelers or 4-6-0 type. The latter is heavier, and has a greater tractive power than the other two, and is intended for what might be called special services.

To these may be added a fourth, the Pacific type (4-6-2). The last, having the same number of drivers as the ten-wheeler, has greater boiler power in proportion to its adhesive weight, and is used in what might be called exceptionally heavy service. The two first classes bear, in a general way, the same relation to each other as the two last, namely, that of having the same number of driving wheels. The Atlantic type has the greater boiler power and is capable of maintaining a higher speed than its predecessor, the eight-wheeled engine. The Atlantic type, first introduced in 1893, possesses so many advantages for heavy and fast passenger service, that it has been rapidly

introduced for that purpose, supplanting the first type in many places where it was originally used.

In order, then, to simplify matters, it will be decided at the outset that the design will be made for the Atlantic type; and, as the work progresses the advantages possessed by the same will be set forth.

With the same weight of rail and the same conditions of track as those set forth in the determination of outlines of the freight engine, we would naturally have the same weight upon the driving wheels. But as there are but two pairs instead of four, the available tractive force drops to 18,250 pounds.

In the case of the passenger engine, there are some complications introduced into the calculation that do not enter into that of the freight engine. The most important one is that of speed. It is evident that ten miles an hour would not at all answer the requirements of passenger service even on the grade mentioned, and the work should be based on a speed of at least thirty-five miles an hour. Owing to the shorter cut-off that will be involved by such a speed, the full adhesive weight of the locomotive cannot be used so that a very liberal reduction will have to be made. The designer knows from analysis that only about 60 per cent of the total adhesive weight is available at such speeds, and as this should be cut down still more to allow a suitable margin for wind and other uncontrollable resistances there is left but little more than 10,000 pounds tractive power that can be used at the speed decided upon.

Referring back to the formula of the *Engineering News* for the train resistance at 35 miles per hour, we find it to be 10.75 pounds per ton, to which should be added the 20 pounds due to grade, making a total of 30.75 pounds per ton. If we divide the available tractive power of 10,000 pounds by 30.75, we obtain 325 tons as the weight of the train, inclusive of the engine, that can be hauled. If the speed were to be dropped to 25 miles per hour, this weight would be raised to something more than 350 tons. It would, therefore, be reasonable specification on the part of the railroad officers to call for a locomotive capable of hauling 350 tons up a 1 per cent grade at a speed of 25 miles an hour, and, if such a specification were to be made, an engine like that shown in Fig. 2, having about 20,000 pounds on each of the driving wheels, and weighing in all 168,000 pounds, would be offered for the service.

CHAPTER II.

THE BOILER.

In the preliminary considerations regarding the designing of the locomotive two main points have been approximately settled: the weight of the engine that can be made to produce the tractive power needed to perform the work that is required and the size of the cylinders that will be needed in order utilize the adhesive weight of the engine, with the steam pressure of 200 pounds per square inch that it has been assumed the boiler is to carry.

It may be remarked here, that in the designing of a modern locomotive all things are made subordinate to the boiler and cylinders, and of these two the boiler is the more important. On it depends the whole action of the engine. If it fails to supply the requisite amount of steam the engine either cannot haul its train without losing time upon the schedule on which it is supposed to run; or, if it supplies the steam, its grate area and heating surfaces may be too small to do the work properly, and the result will be that the engine is extravagant in the use of fuel. For these reasons, then, it is the end and aim of every designer to use as large a boiler as possible in order to obtain an ample supply of steam, and at the same time secure that supply on a minimum fuel consumption. At the same time he is limited by the allowable total weights which must include not only the boiler itself but the cylinders, wheels, axles, machinery and other parts.

In this, as in the work that has already been done, it is impossible to lay down any hard and fast rules, and the designer will frequently find himself thrown back on his own judgment and experience in default of formulated data bearing upon the subject that he has in hand. With this understanding of the matter attention may now be turned to the determination of certain points connected with the boiler of the consolidation freight locomotive that we have in hand, and of which a preliminary outline has been laid down in Figs. 3 and 5.

The two important elements in the boiler are the heating surface and the grate area. The former takes the precedence and is usually based upon scme assumed service that the engine is to render. In the case in hand this has been arbitrarily placed at the hauling of 975 tons as a speed of 10 miles an hour up a 1 per cent grade, or of moving 1,118 tons including the weight of the engine and tender.

For some time the empirical rule for the determination of the amount of heating surface was to make it, in square feet, 400 times the cubic contents of a single cylinder in cubic feet. This rule is, however, only approximately followed and is regarded merely as a

rough guide as to what should be aimed at as a minimum. For, as already stated, it is the desire of the designer to make the heating surface as large as possible, so that this ratio is exceeded wherever it is possible to do so and still keep within the limitations of weights. This is especially true of work in connection with passenger engines, where the demand for steam is apt to be excessive.

Then, too, after the general dimensions of the boiler have been decided upon it is quite possible to vary the heating surface through comparatively wide limits by a variation in the spacing of the tubes. It is in this particular especially that good judgment must be exercised. There is the constant temptation, backed by desire to run the heating surface up, to use a large number of tubes, but it must be borne in mind that it is sometimes advisable to space the tubes more widely apart and put in a smaller number than it is to crowd them; because it is necessary that the steam formed in contact with the lower rows should be free to rise to the surface of the water, otherwise poor evaporation, damp steam or even priming may be the result. It is, therefore, usually better to sacrifice some of the heating surface that it might be possible to obtain, as this will give an actual increase in the evaporative efficiency.

Turning now to the determination of the amount of heating surface and regarding the rule given merely as an approximate guide, it is considered that a more correct basis for the estimate will be to ascertain the weight of steam that will be required to maintain a given speed and tractive power, and from that calculate the amount of heating surface that will be needed to produce it. In short, it is necessary to determine the amount of water that is to be evaporated per minute or per hour, and this, in turn, swings back to the cylinder, where the point of cut-off and the pressure will have to be assumed. This assumption should be based on the records of performances of other engines, and should be critically scrutinized in order to determine the influences that necessary variations in valves, valve motion and steam passages may have upon the result. This means a thorough investigation and the securing of reliable data if a close degree of accuracy is to be obtained.

For the solution of the specific problem that we have before us, the data available have made possible the development of the following formula for the determination of the amount of water to be evaporated per minute by an engine in heavy freight service:

$$W = 2uvnpc \times 1.25 \tag{4}$$

in which

W = pounds of water to be evaporated per minute,
u = the volume in cubic feet of the two cylinders,
v = the percentage of the stroke at which cut-off takes place,
n = the number of revolutions per minute of the driving wheels.
p = the weight of 1 cubic foot of steam at the cut-off pressure,
c = the factor of evaporation from and at a temperature of 212 degrees F.

It will be seen that this formula involves a few points that have

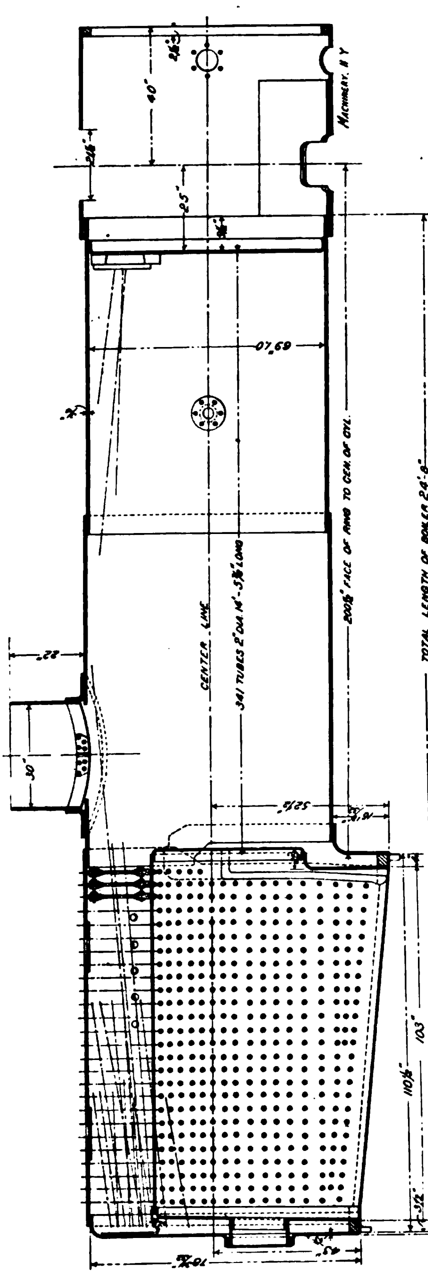


Fig 3. Section of Boiler of Consolidation Freight Locomotive to haul Train of 975 tons up Grade of one per cent. at Ten Miles per Hour.

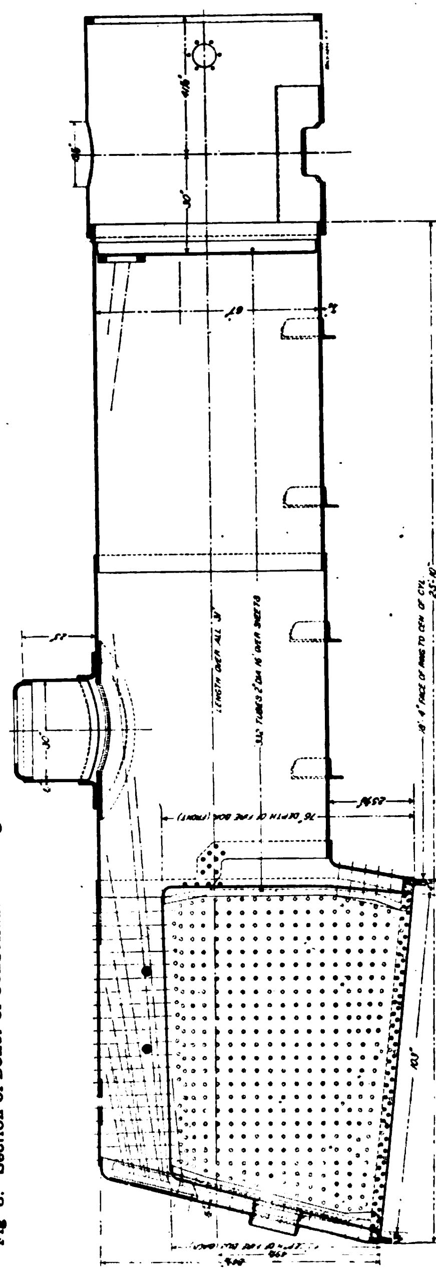


Fig. 4. Section of Boiler of Atlantic Type Passenger Locomotive to haul Train of 350 tons up Grade of one per cent Twenty-five Miles per Hour.

not yet been settled in the course of our work, and which must be decided upon before further progress can be made.

Taking up the equation (4) seriatim, the coefficient 2 is used because the cylinders must be filled with steam twice at each revolution. The factor u is readily determined, and for the case of the cylinders for the consolidation locomotive that are 21 inches in diameter and of 26 inches stroke it is 10.4 cubic feet. For an engine working at the speed and conditions that have been assumed, practical experience shows that the valves may be made to cut off at about 70 per cent of the stroke.

The speed of the engine and the diameter of the drivers are needed in order to determine n. We have already assumed that the speed shall be 10 miles an hour and the diameter of the driving wheels has

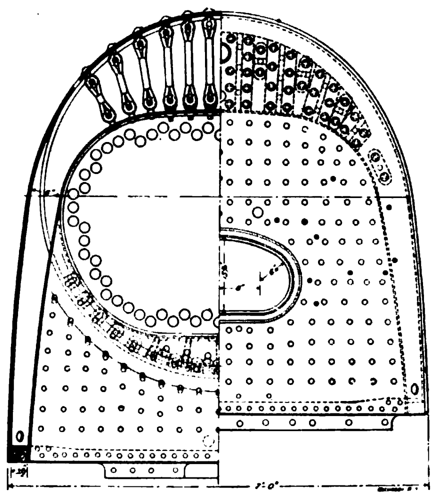


Fig. 5. Cross Section of Boiler shown in Fig. 3.

been put at 57 inches as being well suited to the stroke adopted. These factors taken together give about 60 revolutions per minute, or, more exactly 59.

In ascertaining the weight, p, of the steam at cut-off pressure another assumption must be made. Owing to the slow closing of the valve, and the frictional resistance of the steam pipes and passages, this pressure will be somewhat below that of the boiler, and for the 70 per cent of cut-off at which the engine will be worked at this speed, it has been found to be about 80 per cent of full boiler pressure, or in the case of our assumed pressure of 200 pounds per square inch it is 160 pounds. Referring to the steam tables as ordinarily published we will find that, at this pressure, the weight will be 0.3873 pounds.

The factor of evaporation is obtained by dividing the difference between the heat of the steam at the observed pressure (in this case 200 pounds), and the total heat of the feed water (which is taken at 32 degrees) by the factor of 965.7 which gives in the case in point

about 1.24. The 1.25 at the end of the term is an allowance made for clearances, leakages and the like. With these quantities obtained they may be substituted in the equation (4) with the following result:

$$W = 2 \times 10.4 \times 0.70 \times 60 \times 0.39 \times 1.24 \times 1.25 = 528 \text{ pounds.}$$

That is, the requirements of the engine are such that it must be supplied with the equivalent of 528 pounds of steam per minute from and at 212 degrees F., or 31,680 pounds per hour. The actual evaporation requirement is the amount obtained without the factor 1.25, or 25,350 pounds hourly.

Having determined the amount of work that the boiler will be required to do, the next step is to ascertain the details of dimensions of that boiler to meet the requirements. The first movement in this

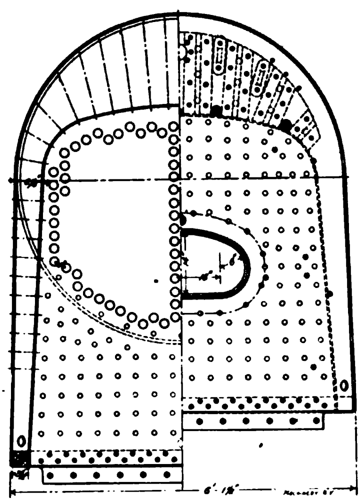


Fig. 6. Cross Section of Boiler shown in Fig. 4.

matter is one that varies with every road and every locality for which an engine can be designed. It involves a determination of the heating capacity of the coal that is to be used. For this reason, when a builder is called upon to design a locomotive whose performance is to be guaranteed he requires a sample of the coal that is to be used in order that its value may be ascertained and the proper proportions of grate area and heating surface be arranged. In the case in hand, it will be necessary to make an assumption and this will be done on the basis of a good quality of bituminous coal, that can be depended upon to evaporate 7½ pounds of water per pound of fuel in a properly proportioned firebox with a suitable ratio between heating surface and grate area.

Experimental examination has shown that, for this grade of coal, one square foot of heating surface is capable of absorbing the heat developed by 1½ pounds of coal burned per hour. We can now em-

ploy the following formula for the determination of the heating surface:

$$S = \frac{W_1}{f\,g} \tag{5}$$

in which

S = the area of heating surface in square feet,
W_1 = pounds of water to be evaporated per hour,
f = pounds of water to be evaporated per pound of coal,
g = the number of pounds of coal allowed per square foot of heating surface per hour.

Substituting the values already obtained the formula becomes,

$$S = \frac{31{,}680}{7.5 \times 1.5} = 2{,}816$$

The grate area may be determined by the substitution, in equation (5), for g a factor expressing the amount of coal to be burned per square foot of grate per hour. The equation then becomes:

$$G = \frac{W_1}{f\,h} \tag{6}$$

in which

G = the grate area in square feet,
h = the amount of coal in pounds to be burned per square foot of grate area per hour.

It is common practice in the freight service of American locomotives to burn 100 or more pounds of coal per square foot of grate per hour. This means that the fires are to be forced to a greater or less extent. and the conservative designer who wishes to make sure that his engine will meet all of the requirements of the specifications will assume a much lower rate of combustion than this, in order to have a reserve power for use in cases of emergencies. It will be well, then, to take 75 pounds per square foot per hour as the rate of combustion. By substituting this and the other values that have been obtained in equation (6) it becomes

$$G = \frac{31{,}680}{7.5 \times 75} = 56.3, \text{ or } 57$$

The boiler called for should therefore have 2,816 square feet of heating surface, and a grate area of 57 square feet. On comparing the ratio of grate to heating surface obtained in this way it will be found to be as 1 to 50, which is below the average of construction. It must be borne in mind that these are not hard and fast dimensions, but may be varied according to the exigencies of the case, the limitations of construction and the modifications suggested by the experience of the designer. The distribution of the heating surface and the determination of the weight of the boiler is the next step.

In this, much depends upon the type of the boiler, the wheelbase and other features of general construction. These limit or indicate the length of the tubes. This may range anywhere from

13 feet to 20 feet, although about 15 feet is now an approved dimension. In a general way it may be said that the longer the tube the more heat is extracted from the gases of combustion, though, it has been found that if the heating surface remains the same the length of the tube is a matter of indifference within certain limits. The reason for this is that with a long tube the gases move rapidly but remain in it for the same length of time as in a short tube where they move slowly, there being more short than long tubes for the same heating surface. As for the weight of the boiler this may be taken to average from 30 to 33 per cent of the total weight of the engine.

The type of the boiler is usually settled by preference and past experience with a limiting factor of present requirements. The general types of boilers are, broadly speaking, the straight and wagon-top. In either of these the crown of the firebox may be stayed with crown-bars or radial stays, or be of the Belpaire type, and the firebox itself may be broad or narrow; that is, it may be extended over the wheels, set on top of the frames or drop down between them. The straight-top boiler is the more common of the two first divisions because of the inconvenience of utilizing the wagon-top on large modern engines, where everything is set well up, on account of the limitations of the overhead clearance of the permanent way.

Of the methods of staying the crown-sheet, the crown-bars have been practically discarded. The Belpaire type has not been widely adopted, while the use of radial stays is most widespread. Finally the wide firebox set above the wheels is a necessity for large engines in order to obtain the large grate area required in modern motive power and the narrow-construction between the frames may be regarded as a thing of the past. So in the boilers for the engines under construction, the straight-top type, with a wide firebox held by radial stays will be taken as best suited to the requirements of this case.

Referring back to the general outline of the consolidation engine of Fig. 1, it will be seen that the wheels are spaced rather near together. The object of this is to secure as short and rigid a wheelbase as possible. It is evidently out of the question to bring the wheels so near together that there is a bare clearance between the flanges, on account of the necessary attachments of the frames. It has been found, however, that from 5 inches to 7 inches between the treads will answer, so that in this engine, with wheels 57 inches in diameter, the distances between centers are taken at 64 inches, 62 inches and 64 inches respectively.

With a cylinder having a piston stroke of 26 inches the total length of the same will be about 36 inches. Allowing necessary clearance between the tread of the forward driving wheels and the cylinder casting, the distance from the center of the forward driving wheel to the center of the cylinder may be taken as the case requires. As the boiler proper usually ends at the cylinder casting, in this case it will be 42 inches ahead of the forward center. Finally in order that the boiler and engine may balance well on the wheels and not tend to

put an excessive load on the rear drivers or front truck, the overhang of the firebox back of the rear drivers should be made with due consideration to the position of the wheels and distribution of the weight. On this basis the approximate length of the boiler from the back head to the front tube sheet would be about 24 feet 6 inches or 104 inches more than the wheelbase. Now comes the proper distribution of this distance into firebox and tube lengths.

The requirements of the service will demand that the firebox extend out over the wheels, so that constructional limitations will decide the total width. With a firebox tapering in at the top, a total width of about 7 feet at the mud-ring can be used. Allowing for the thickness of metal and a width of ring on each side of 3½ inches will leave about 75¼ inches, or 6¼ feet, for the width of the grate. In order to obtain 57 square feet of grate area the length should be a little more than 9 feet, and deducting 4 inches at the rear for the water leg, would leave a length of about 15 feet for the tubes.

As already stated, these dimensions are not fixed, so that in the designing of the boiler some modifications are possible, to suit the requirements and conveniences of construction. If the work were to be undertaken from the start a great many trials would have to be made on the drawing board in order to secure the proper adjustments. Without reviewing these steps one by one, it will be permissible to state that if a firebox of the width given be used, and it be made 6 feet high above the mud-ring, it will have a heating surface of about 15½ square feet per foot of length in addition to about 54 square feet for the ends, if no allowance is made for tube and door openings. Suppose, then, this firebox be made 8 feet 6 inches long inside; the heating surface will be 186 square feet, leaving 2,630 to be made up in the tubes. With 0.5 square feet per lineal foot as the approximate heating surface of a 2-inch tube, this would require a total tube length of 5,260 feet. As the available length is 15 feet, this would require 351 tubes.

With this as a guide, it will be found, in laying out the best form cf firebox, and allowing the space needed above it for water and steam, and by keeping within the limits imposed, that 341 tubes 2 inches in diameter will be the most convenient number and that best adapted for service. They will have a length of 14 feet 6 inches over the tube-sheets and the inside diameter of the smallest ring of the shell will be 69 inches, all of which is shown in the general outline of Fig. 3. On taking this boiler and calculating the actual heating surface it will be found that there are 2,570 square feet in the tubes to which the 186 square feet in the firebox is to be added, making 2 756 square feet in all.

Without entering into the details of the proportioning of the parts in order to secure the proper strength, which is really the next step, but which is outside the province of this work, attention is to be turned to making an estimate of the weight of such a boiler as that outlined. It will be found that the weight of the materials in the boiler itself with its firebox and bracing complete, but without the

tubes will be about 30,000 pounds to which must be added the weight of the latter or 11,000 pounds, thus bringing the total weight up to 41,000 pounds.

With the weight and dimensions thus established, it will be possible to go on and work out the other parts of the mechanism.

Turning now to the passenger locomotive, the boiler must be worked out somewhat differently. Here we are not confronted with the hauling of a very heavy load at a low speed, but with the work to be done at a comparatively higher speed with a lighter load. This will involve the working of the engine at a shorter cut-off and here again experience in the working of an engine will be called into play.

In the preliminary considerations of this matter the requirements were laid down that the engine should be capable of hauling a passenger train weighing 350 tons up a grade of 1 per cent at a speed of 25 miles an hour. The engine selected for this purpose was of the Atlantic type, as being that best adapted to the work. As in the case of the consolidation locomotive the diameter of the driving wheels must be arbitrarily determined. In this the designer is to be guided by the necessity of obtaining a reasonably good speed on lighter grades and level; and for this a diameter of 77 inches will be found to be well suited.

From the formulas (1) and (2) it will be found that the resistance of the train will be $R = \frac{25}{4} + 2 = 8.25$ pounds, and $R' = 2{,}000 \times \frac{1}{100} = 20$ pounds, or that the total will be 28.25 pounds per ton, and $350 \times 28.25 = 9{,}888$ pounds, or that, in round numbers, a drawbar pull of 10,000 pounds will be required to do the work.

It is evident that the varying resistances of the train do not become any the less when there is an increase of speed, but rather increase. On the other hand, the power of the engine diminishes with the increase of speed. So instead of the 11 per cent margin allowed for uncontrollable resistance, in the case of the freight locomotive, an allowance of at least 15 per cent must be made for the passenger locomotive. By allowing a 15 per cent increase of resistance and a similar loss of power in connection with the needed drawbar pull we have $\frac{10{,}000 \times 0.15}{1 - 0.15} + 10{,}000 = 1{,}765 + 10{,}000 = 11{,}765$ pounds. Further, the internal friction of the engine is not reduced in proportion to the fall in the mean effective pressure in the cylinders. So instead of taking this at 10 per cent, as in the case of the consolidation, it has been found by tests and experience that an allowance of 18 per cent of the theoretical tractive power must be made. Hence the minimum tractive power needed would be $11{,}765 \times 1.18 = 13{,}882.7$ pounds, or in round numbers, 13,900 pounds.

For the determination of the piston speed, to be used in calculating the proper diameter, the following formula may be used:

$$S = \frac{12 \times 5280\, v \times 2\, p}{3.1416\, D \times 60 \times 12} \qquad (7)$$

in which

S = the piston speed in feet per minute,
v = speed of the train in miles per hour,
D = diameter of the driving wheels in inches,
p = the stroke of the piston in inches.

In order to fill in the required factors in this case, it is necessary to assume a piston stroke that will be well adapted for use with the wheel diameters that have been given and the service to be performed. This will be placed at 26 inches. Then by substitution the equation (7) becomes:

$$S = \frac{12 \times 5280 \times 25 \times 2 \times 26}{3.1416 \times 77 \times 60 \times 12} = 473 \text{ feet per minute.}$$

Indicator diagrams have shown that at a piston speed of 500 feet per minute the mean effective pressure in the cylinder will be about 62 per cent of the boiler pressure.

By substituting the values which have now been obtained in equation (3) it is possible to obtain the diameter of the cylinder. This is then:

$$d = \sqrt{\frac{18{,}900 \times 77}{200 \times 0.62 \times 26}} = 18.5 \text{ inches, nearly.}$$

In order that there may be an ample margin of power the cylinder diameter, in this case, will be increased to 19½ inches, as the weight allowed will make it possible to use a boiler of sufficient capacity to supply such a cylinder.

The boiler dimensions may now be determined in the same way as for the freight engine by the substitution of the several values in equation (4), for the calculation of the equivalent amount of water to be evaporated per minute. This then becomes:

$W = 2 \times 9 \times 0.55 \times 110 \times 0.30 \times 1.24 \times 1.25 = 506.4$, or 500 pounds per minute, or 30,000 pounds per hour.

Taking the same quality of coal as before, and allowing 1½ pounds of coal per square foot of heating surface per hour, the latter, according to formula (5) should be:

$$\frac{30{,}000}{7.5 \times 1.5} = 2{,}666 \text{ square feet,}$$

and from formula (6), with a rate of combustion of 85 pounds per square foot per hour of grate area, the latter will be:

$$G = \frac{30{,}000}{7.5 \times 85} = 47 \text{ square feet.}$$

Proceeding as before, it will be found that the most convenient dimensions for tubes and surfaces will give a boiler with 2,767 square

feet of heating surface in the tubes and 162 square feet in the firebox, making a total of 2,929 square feet and a grate area of 46.3 square feet.

This is somewhat in excess of that called for by the formula in the way of heating surface, but it will be found to be of great advantage in a passenger engine and can only be obtained in this type of locomotive, though there is some objection to the extra weight on the forward truck and trailing wheels.

The length of the firebox will be 102 inches by 65½ inches wide. There will be 332 tubes of 2 inches diameter, 16 feet long, and the diameter of the smallest ring of the shell will be 67 inches. Finally, the weight of the boiler will be 39,300 pounds.

In the matter of the evaporative qualities of the various grades of coal and the consumption per hour the following table is presented as deduced from a report made to the American Railway Master Mechanics' Association in 1897:

	Water Evaporated per lb. of Coal.	Coal to be Burned per sq. ft. of Grate Area per hour.
Large Pennsylvania anthracite	8 lbs.	60 lbs.
Fine Pennsylvania anthracite	6½ lbs.	35 lbs.
Virginia semi-bituminous	9 lbs.	65 lbs.
Illinois bituminous	7 lbs.	90 lbs.

Finally the following ratios were suggested:

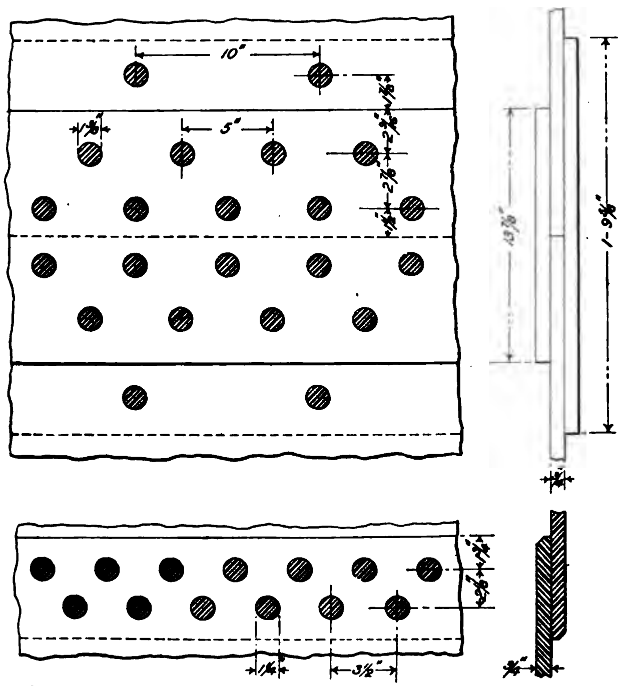
	Cyl. Volume in cubic feet to Grate Area in square feet.	Cyl. Volume in cubic feet to Heating Surface in square feet.	Heating Surface to Grate Area.
Large anthracite	1:4	1:180	40:1
Small anthracite	1:9	1:200	20:1
Bituminous	1:3	1:200	60:1

In the work that has preceded, the various steps leading up to the determination of the general dimensions and proportions of the boilers have been indicated. With this accomplished there still remains a great deal of work to be done in laying out the details of the several parts of the boiler itself. While it will be impossible to enter into a discussion of boiler construction in detail the importance of this part of the locomotive is such that some attention should be paid to it here.

It has already been remarked that the boiler is the life of the locomotive, and that upon it depends the efficiency of the whole machine. Hence it is of the utmost importance that the greatest care should be exercised in its design and construction to make sure that it is possessed of the requisite strength and power of endurance.

In this, attention is first turned to the shell whose strength depends upon the thickness of the plate of which it is formed and the type of longitudinal seam used to connect the edges of the same. In regard to the latter, that one should be selected that will give the highest percentage of strength, as compared with that of the solid plate, consistent with practicability of construction and maintenance in service.

Without entering into a discussion of the relative merits of the various types of joints it may be stated that the sextuple riveted, double butt strap joint with welts inside and outside the main plates as shown in Fig. 7, is best adapted for this work and will, therefore, be used. In this joint there are three rows of rivets on each side of the joint, all of which pass through the inner strap, and but two through the outer. For an analysis of the strength of such a joint, the reader is referred to the various handbooks on boiler construction, where it will be found that the calculated strength of such a seam is



Figs. 7 and 8. Sextuple Riveted Butt Strap Seam, and Double Riveted Lap Seam.

a little more than 86 per cent of that of the solid plate and that 85 per cent can be counted upon in regular working practice and construction.

With this as a preliminary basis, the strength and thickness of the shell can be calculated from the following formula:

$$L = \frac{S\,C}{f} \tag{8}$$

in which

L = the working stress that can be put upon the plate per square inch of section,

S = the ultimate tensile strength of the steel in pounds per square inch,

C = the percentage of strength of the seams of the plate,

f = the factor of safety that it is necessary to allow.

The specifications for boiler steel of the American Railway Master Mechanics' Association state that "the desired tensile strength is 60,000 pounds per square inch, with minimum and maximum limits of 55,000 and 65,000 pounds." As, in the design it will be necessary to work to the minimum, 55,000 pounds per square inch will be taken as the tensile strength. Practice has further shown that a factor of safety of 5 is well adapted to working conditions. Hence by the substitution of the values obtained in the second term of the equation we have

$$L = \frac{55,000}{5} = 11,000 \text{ pounds}$$

per square inch when referred to the solid portion of the plate, although when referred to the efficiency area of the seam this is

$$L = \frac{55,000 \times 0.85}{5} = 9,350 \text{ pounds.}$$

The actual thickness of the shell plate is calculated by the formula

$$T = \frac{D' P}{2 L} \qquad (9)$$

in which

T = the thickness of the plate in inches,

P = the working pressure,

D' = the diameter or the boiler in inches.

The coefficient 2 is in the denominator to allow for the stress carried by two sheets, at opposite ends of the diameter. Referring to the boiler intended for the consolidation engine, equation (9) becomes, by substitution,

$$T = \frac{69 \times 200}{2 \times 9,350} = 0.738.$$

We therefore take ¾ inch as the proper thickness of the smallest ring of the boiler shell.

In the case of the boiler for the Atlantic type engine, the inside diameter is 67 inches and the equation (9) becomes

$$T = \frac{67 \times 200}{2 \times 9,350} = 0.716.$$

Owing to the liberal factor of safety that has been adopted, it will be allowable to take the sheet in nearest sixteenths which will give 11-16 inch as the thickness of the smallest ring.

In these calculations the rings of the sheet have been considered as though they were integral and unbroken. This is true of the front sheeet but in the second, the opening for the dome weakens it to such an extent that it is generally made about 1-16 inch thicker than that calculated. Hence, in the case of the consolidation locomotive this sheet should be 13-16 inch thick, while for the Atlantic ¾ inch

may be used, although this falls a little short of the extra 1-16 inch called for.

The dome opening should be made as small as possible, besides being thoroughly braced by a heavy dome base and an inside liner around it, as clearly shown in Fig. 4 of the passenger engine boiler. This opening should be limited to that actually needed for the entrance and adjustment of the standpipe, and for its use as a manhole—a rule that applies equally well to the diameter of the dome. The idea that the dome can serve any useful purpose as a storage reservoir for steam has long since been discarded as it is merely a means of elevating the throttle above the water and thus securing dry steam for the cylinders.

Although the neutral part of the shell within the lines of rivets holding the dome in place should not be taken into consideration in calculating the strength of the former, it nevertheless does serve a very useful purpose, and materially adds to the strength of this part, in that the flexible edge around the opening transfers the stress from the edge to a line inside the inner row of rivets of the dome base, and thus to the solid portion of the shell, and thereby lessens the tearing effect that would exist if the opening were cut out so as to leave only the usual margin inside the rivets.

In the designing of the circumferential seams, the steam pressure may be disregarded, and the work done with consideration only to tightness and structural strength. A brief consideration of the stresses to which the boiler is subjected will show that, when the tubes are in position they, together with the bracing of the front and back heads, so relieve the longitudinal seams from the stresses due to seam pressure alone that the latter becomes a negligible quantity. For that reason a double riveted lap seam is used which, while it has perhaps less than 70 per cent the strength of the solid plate, is ample for its work. Such a seam is shown in Fig. 8.

Turning now to the firebox, the inside sheet should be made thin so as to offer the minimum resistance to the tranmission of the heat of the fire to the water beyond. The proper spacing of the staybolts will, of course, make it possible to use almost any thickness of sheet; hence the choice must be made as the result of experience rather than from any mathematical calculations. This experience has shown that when the steam pressure ranges from 180 pounds to 200 pounds per square inch, the best results can be obtained with side sheets ⅜ inch thick. This does not hold for the tube sheets, where, on account of the necessity of expanding and fixing the tubes, the thickness should never be less than ½ inch and the use of ⅝ inch will frequently be found advisable. So, basing the choice on this general principle, the thickness of the metal in the side and tube sheets of the two boilers will be taken at ⅜ inch and ½ inch respectively, the crown-sheets being given the same thickness as the sides.

The general practice at the present time for supporting the crown-sheets is by means of radial stays, while the flat surfaces are held by the ordinary staybolts. In some cases the latter are replaced at

the upper forward portions of the firebox, where there is the greatest difference between the expansion of the inner and outer sheets, by flexible staybolts of various designs. Owing to this difference in the expansion of the two sheets, and the bending of the staybolts resulting therefrom, it is desirable that the latter should be made of a comparatively small diameter. In practice, on boilers carrying a high steam pressure, the diameters range from ⅞ inch to 1⅛ inch as a maximum, and they are usually spaced so that the stress put upon them does not exceed 5,500 pounds per square inch of section. The spacing runs from 3¾ to 4½ inches from center to center of the bolts. If, then, 4 inches is adopted in the boilers under consideration, the total stress on the 16 square inches held by each bolt will be 3,200 pounds, which will call for a staybolt ⅞ inch diameter and load it to about 5,330 pounds per square inch of section.

To provide for the injurious stresses imparted on the flue-sheet and shell by the expansion of the former in advance of the latter in raising steam in the boiler, sling stays of a telescopic nature are applied over the forward part of the firebox so that the latter is allowed to rise slightly until the shell of the boiler is heated by the water and steam, which then expands and gradually takes up the slack thus formed and brings the stays under tension as the temperature and pressure increases.

The width and shape of the water legs has a most important bearing on the efficiency and durability of the boiler. These two points should be so related to each other that there is room for the inflowing current of water and the escaping steam that is generated in contact with the inner sheet. Carefully conducted experiments have shown that, in some cases, where the sheet is vertical there is hardly any water in contact with the upper portions, but that they are covered with a layer of steam bubbles ascending to the surface. Such a condition has the double disadvantage of lowering the rate of evaporation and leading to an overheating of the sheets. In order to avoid this, the sheet should be so sloped in that the steam, in rising vertically, tends to leave it and give free access to the water. It will be noticed from the cross-sections in Figs. 5 and 6 that this has been done in both boilers.

Again. owing to the fact of there being more steam at the upper portion of the leg than the lower, it should be widened at the top, and this has also been done in both cases. At the mud-ring the width used is 3½ inches while at the top 6 inches is required. Such a space will be found to work well in practice, and is founded on sound reasoning rather than mathematical calculations. There are many other points of more or less importance that could be referred to, but available space requires that the discussion should be limited, in the main, to general principles. Finally a few recommendations should always be borne in mind, and the work designed in accordance therewith. Among these, one of the first importance is that flat surfaces should be avoided as far as possible, and where they cannot be done away with entirely, they should be reduced to the lowest dimen-

sions, so as to cut down staybolt stresses to a minimum. Wherever flanging or bending of the sheets is required, it should be done with a liberal radius of curvature, so as to secure flexibility and avoid undue internal stress of the material. In the application of the bracing great care should be exercised that it gives strength without adding too much to the rigidity. A boiler is subjected to such varying temperatures in its different parts that it must expand and contract differently; hence it is of the utmost importance that it should be

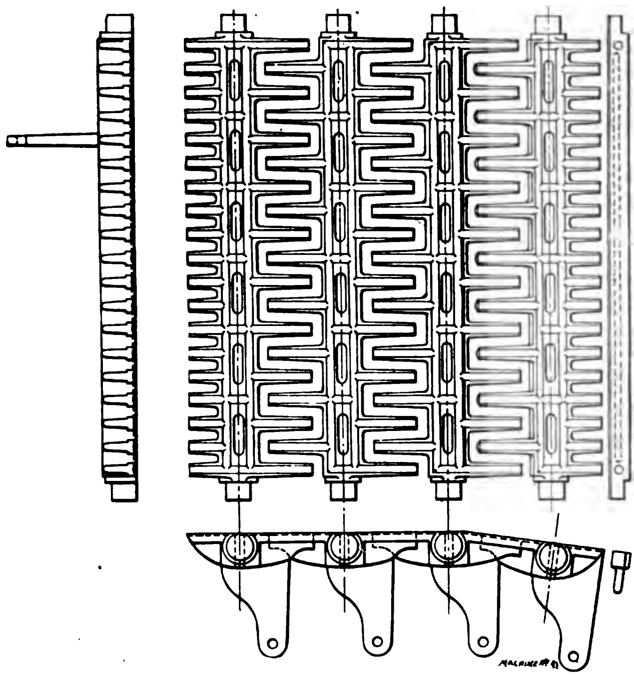


Fig. 9. Grate for Bituminous Coal.

capable of this internal movement of its parts relatively to each other, without putting an undue stress upon the metal of which they are composed.

A few words may be added regarding the actual work of constructing the boiler, for this, while not strictly belonging to the work of the designer, should, nevertheless, be borne in mind by him, and may well form a part of his specifications. Without taking up the matter in all its detail, a few points will be touched upon.

Next to the formation of a tight joint at the sheets, it is of the utmost importance that the tubes should be well and efficiently set. To do this it is advisable to make the holes in the front tube-sheet 1-16 inch larger than the diameter of the tubes, while those at the firebox end should be of the exact size. Both edges of the holes should be chamfered to a radius of 1-16 inch. Further, a copper

ferrule should be used at the firebox end, and this should have the same outside diameter as the tubes, a thickness of about 1-16 inch and a length ¼ inch greater than the thickness of the sheet. The ferrules should be lightly rolled and expanded in place before the tubes are inserted, while these should be swaged down so that they will just enter the ferrules. At the front end, on the other hand, they should be expanded and rolled out to fill the holes, care being taken that the ends are annealed before they are put in place. As to length, the tubes should be cut so that they will project from ⅛ to 3-16 inch beyond the sheets at each end. A satisfactory system of tube-setting is to turn back about 10 per cent of the tubes at each end to be beaded over in order to act as stay tubes, and then to round up the edges of all the tubes with a mandrel. Of course it goes without saying that the rolling of the tubes should be carefully done, and if necessary, should be repeated at the firebox end after the boiler has been fired and tested.

Turning back to summarize the stresses that have been given as allowable for staybolts and other related and similar parts, experience has shown that for those in the side, front and back water legs the load should not exceed 5,500 pounds per square inch of section. For radial stays that are longer and are not subjected to such excessive bending stresses, due to the variation in the expansion of the two plates that they connect, the load may be raised to 8,000 pounds per square inch, while on the other bracing on which there is little or no bending stress, a working load of 10,000 pounds per square inch may be imposed.

In the firebox the common American practice on all engines burning bituminous coal is to use the finger-grate type of bar having openings between each bar of from ¾ inch to ⅞ inch in width. Such a grate is shown in Fig. 9. As for the brick arch it may be considered to be practically out of use on all wide firebox engines like those which we are now considering in detail. For the narrow firebox type of boiler, however, the brick arch offers a very important advantage. The reason for this variation of practice in the two types is due to the fact that the movement of gases in the large box is slower and combustion more perfect than it would be in the narrow construction, if this latter be without the assistance of the arch for mingling and maintaining a high temperature of the gases.

CHAPTER III.

THE CYLINDERS.

Closely allied with the boiler in importance are the cylinders. As the boiler is the important element in converting the potential energy of the fuel into that of the steam, so the cylinders serve as the means of converting this potential into dynamic energy and thus produce the useful work for which the machine, as a whole, is designed. The matter of the size of the cylinders has already been considered, in the determination of the general dimensions of the engine, where it was found that a diameter of 21 inches, and a piston stroke of 26 inches would be suited to the work that it is intended that the freight or consolidation engine should perform. At the same time the diameter of the cylinder and stroke of the piston of the passenger engine were calculated to be 19½ inches and 26 inches respectively. The cylinders are invariably made of cast iron, and it is of great importance that the metal should be of a character suitable for the work that it has to perform. It should be of a fine grain and as hard as can be worked, the latter quality being needed in order that it may withstand the hard wear of the pistons and valves. A common form of specification is to require that the "cylinders shall be made of a hard, compact, tough iron, of not less than 25,000 pounds tensile strength per square inch, and so cored as to produce uniform shrinkage in cooling." A metal that will meet these requirements can be made from 50 per cent pig, 25 per cent of old carwheels, and 25 per cent of high grade machinery scrap.

The practice of the several builders varied somewhat in the past in the matter of the form of the cylinder and saddle. At one time the saddle was made a separate casting, with the cylinders bolted on outside the frames. In this case the steam and exhaust connections to the smokebox were made either direct from the cylinder casting or from the side of the steam chest. This practice was succeeded by the almost universal adoption of the cylinder and half saddle cast in one piece as shown in Figs. 10 and 11. It will be observed in the two cylinders here illustrated, that Fig. 10 is adapted to the use of a flat slide valve, while Fig. 11 is fitted with a bore for a piston valve, the former being intended for use on a consolidation freight locomotive and the latter on the Atlantic passenger engine. The reasons for this variation in practice will be explained later. The length of the cylinder is dependent upon four factors: the stroke, the thickness of the piston, the clearances at the end, and the amount of counterbore allowed for the inset of the cylinder heads. The length of the working barrel of the cylinder is usually made equal to the stroke of the piston plus the width over the piston packing rings less ¼ inch. This last

subtraction is made so that the piston rings will travel beyond the edge of the counterbore at each end by ⅛ inch and thus prevent the formation of shoulders at the end of the stroke due to the wear of the cylinder barrel. This, of course, involves the determination of the widths of the packing rings, their number and the spaces between them, but this will be referred to later, it being sufficient to state

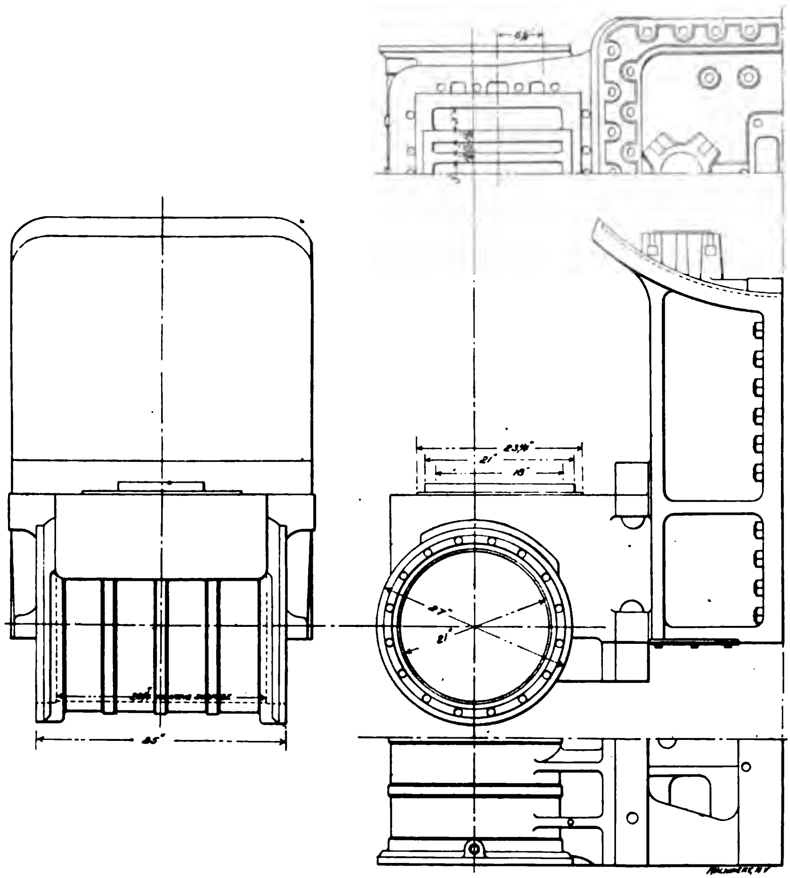


Fig. 10. Cylinder and Half Saddle for Consolidation Locomotive.

here that the width of the pistons over the packing rings, in the case of those used in the two cylinders here shown is 3½ inches, which, with the stroke of 26 inches and the subtraction of ¼ inch, will make the bore of the cylinder 29¼ inches long in each case. The inner edge of the ports should come nearly opposite the outer edge of the packing rings at the end of the stroke, which places them 29¾ inches apart. The ends of the cylinders outside the working barrel are usually bored from ½ inch to 1 inch larger in diameter than the barrel itself to allow for wear and reboring without interfering with the fit of

the cylinder heads. Sometimes, as in the case of Fig. 11, the cylinder is bored through from end to end to the full diameter of the counterbore and a bushing is inserted which can be renewed when it is worn, without disturbing or changing the dimensions of the pistons and packings.

As already indicated, the steam ports and cylinder heads enter the counterbore at the ends which usually extend from 4 to 5 inches beyond the actual stroke of the piston. In Fig. 10, this extension is 4½ inches, making the total length of the cylinder 35 inches, while in Fig. 11 it is 5 inches, making that cylinder 36 inches long. Subtracting from this distance the total thickness of the pistons (5¼ inches for the freight and 5½ inches for the passenger locomotive), we will have 3¾ inches and 4½ inches respectively for the clearances and the counterbore for the heads. It is desirable that the clearances should be as small as possible, though it is necessary that they are large enough to avoid all possibility of the pistons striking the heads. This will frequently be sufficient to take care of the port and steam requirements when the possible variations of motion due to faulty workmanship, wear, and changes effected by the keying of the rods, are guarded against. At times, however, this is not the case, since it has frequently been found to be desirable to give a high-speed engine a somewhat greater clearance than a slow one because of the longer period during which the steam is worked expansively, followed by an earlier compression. The cylinder heads should be so designed and proportioned that they will extend into the counterbore of the cylinder sufficiently for the clearance left between their inner faces and the piston, when the latter is at the extreme end of the stroke to be not less than ¼ inch nor more than ⅜ inch. This clearance should be so divided that when the new engine is turned out of the shop there will be from 1-16 inch to ⅛ inch more clearance allowed at that end of the cylinder toward which the wear of the rod brasses has a tendency to draw the piston than at the opposite end. As to whether this will be the front or the back will depend on the details of the ends of the main rods.

The design of the cylinders is so closely allied to that of the valve motion, and so much depends upon the proper distribution and flow of the steam, that the size and arrangement of the ports will be considered in connection with the valves. In the actual construction of the cylinder, however, the ports as they enter the barrel are usually about 2 inches shorter than the diameter of the bore, increasing somewhat in length at the valve in flat slide valve engines, so that they are a trifle longer or about the same length as the nominal diameter of the cylinder. In the case of the piston valve the length of the port as it enters the cylinders is of about the same proportion as before, but is narrowed as it approaches the steam chest on account of the diameter of the latter being less than that of the cylinder. Here the steam ports extend entirely around the steam chest, being interrupted at a few points for the insertion of bridges to carry the packing rings of the valve over the openings. In laying out the

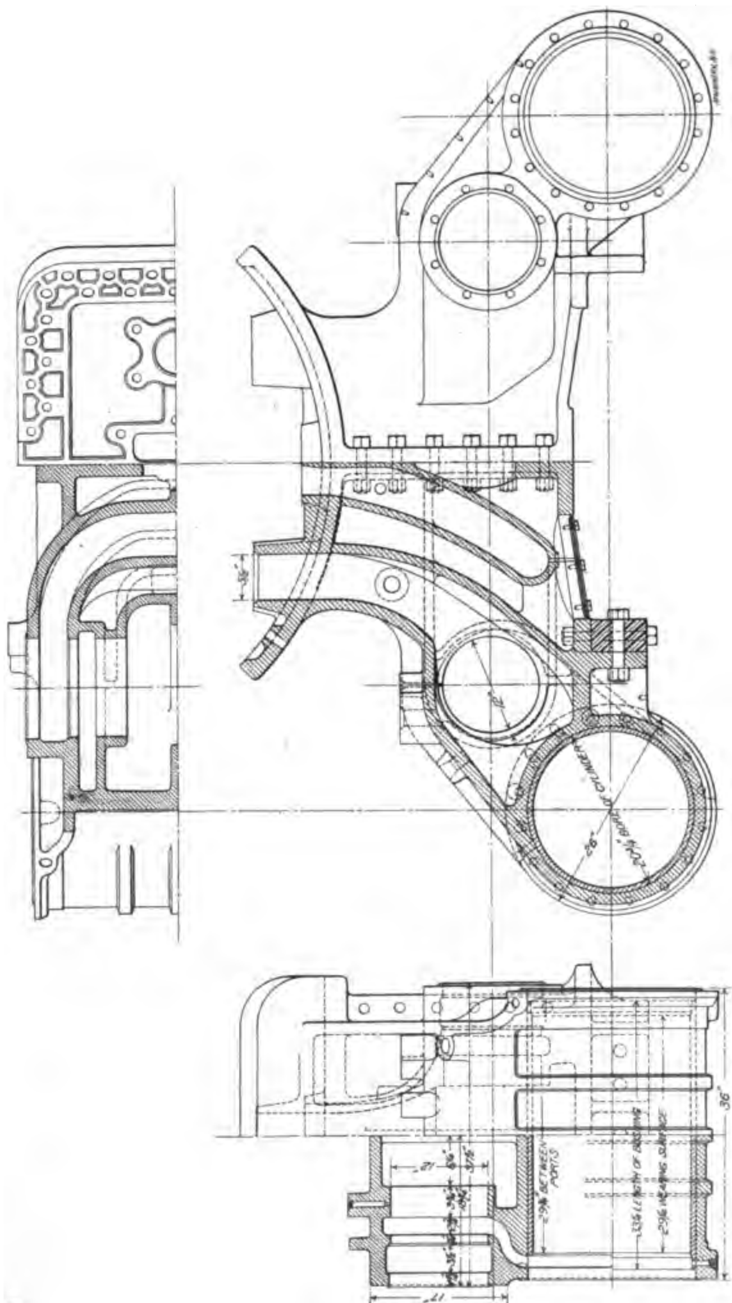


Fig. 11. Cylinders and Saddle of Atlantic Type Locomotive.

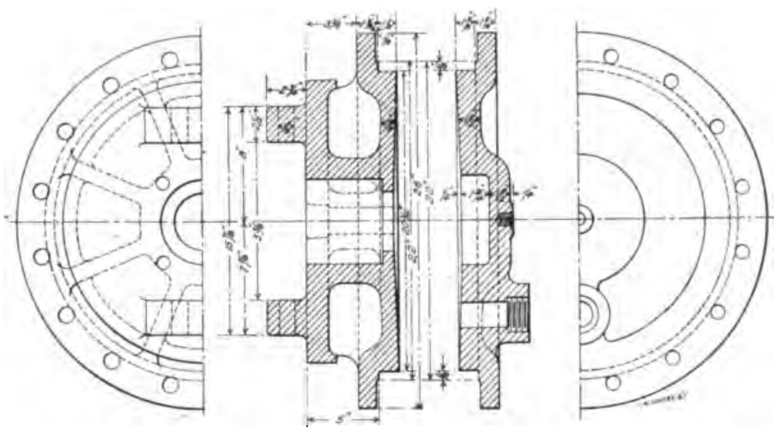
course of the ports care should be taken that they are of a uniform section, as short and direct as possible, well rounded on all curves and turns, and with the turns of as long a radius as possible. These are matters of the utmost importance, since it is essential, for an efficient action of the machine, that the flow of the steam from the valve to the cylinder should be free and unimpeded, and devoid of eddies and cross currents which will tend to reduce the speed of such flow. This is especially important in the case of high-speed engines where any checking of movement of the steam will have a very marked effect on the work that can be done. Where the ports widen out in the saddle to meet the steam and exhaust pipes, they should be clean and roomy, and this is especially true of the exhaust ports. It has been found to be fully as difficult, if not more so, to get rid of the steam after it has done its work in a high-speed locomotive as it is to get it into the cylinder in the first place; hence the exhaust passages should be designed with ample spaces and easy curves that there may be the minimum amount of resistance set up to the escape of the slow pressure steam found in the cylinder at the end of the stroke. In short, every effort should be made to reduce back pressure to the lowest point.

The steam passages should be protected in every possible way from radiation. Under no circumstances should they be allowed to pass along the outer walls of the saddle, but should always be protected by an air space insulating them from the cold outer air. It is well, too, to add to this the further protection of some non-conducting material. Not only should these air spaces separate the steam passages from the external walls, but they should be so arranged as to isolate them, whenever practicable, from the exhaust passages wherein the temperature is much below that of the steam at boiler pressure. In fact, the end and aim should be to deliver as many heat units as possible to the cylinder, for it is upon this that the efficiency of the engine largely depends. In addition to this care by insulation, the steam passages should be laid out in an elastic curve, so that when their walls are heated and cooled by the admission and withdrawal of the steam, the expansion of the metal may not be such as to put any additional stress upon the other parts of the casting. This is especially necessary at that portion of the saddle near the frame fits which must be ribbed in order to withstand the working stresses.

As for the thicknesses of metal to be used in the saddle, there are no reliable formulas available whereby a mathematical calculation of this can be made. Experience has shown, however, that too great a thickness is detrimental to the securing of the greatest strength. This is particularly true where there is an abrupt change in thickness, as the casting is apt to crack at such a point. All outside ribs should be avoided, since they have the double disadvantage of being liable to fracture and serving as a good radiating medium for such heat as the saddle necessarily takes up from the steam. Furthermore, the corners should be well rounded so that both the external and internal stresses may be evenly distributed over the large surfaces. The flanges for

bolting the cylinders and half-saddles together and to the boiler must be made heavy and be ribbed so as to withstand the pressure and driving of the bolts. Here, too, the metal should be arranged so as to join the walls with curves of long radius and large fillets. The walls should be well braced internally with ribs so as not to yield at the root of the flanges. No formula is available for calculating the thickness of these ribs since no once knows exactly, or even approximately, the stresses to which they are subjected. They are ordinarily made from 2½ inches to 3 inches thick, which, in the cases before us, is taken at 2¾ inches.

About the only point in the design of the cylinder that is subjected



Figs. 12 and 13. Back and Front Cylinder Heads for Atlantic Type Locomotive.

to a mathematical analysis is the thickness of the shell of the barrel, which may be calculated from the formula

$$t = \frac{dp}{2S} \qquad (10)$$

in which

t = thickness of the shell in inches,
d = diameter of the bore in inches,
p = steam pressure in pounds per square inch,
S = safe stress in metal in pounds per square inch.

Allowing 2,500 pounds per square inch as a safe working stress for the metal, and substituting the values that have been already found for the two engines, the formula becomes

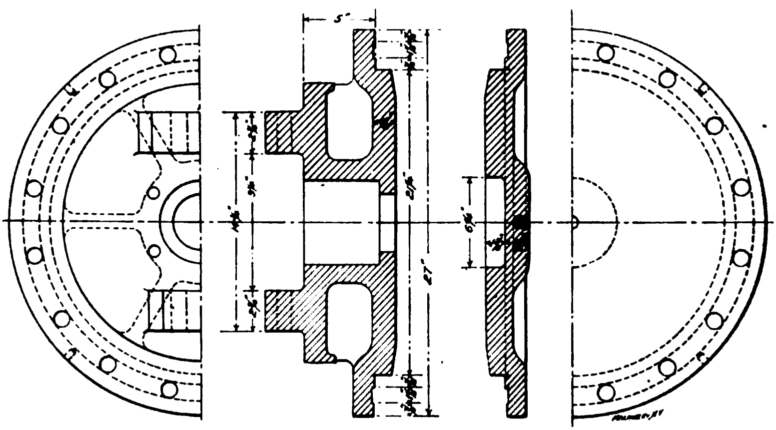
$$t = \frac{21 \times 200}{2 \times 2{,}500} = 0.84 \text{ inch}$$

for the consolidation engine, and

$$t = \frac{19.5 \times 200}{2 \times 2{,}500} = 0.78 \text{ inch}$$

for the Atlantic type. Unfortunately the resistance to the pressure of the steam is not all that is required of a cylinder. It is subjected to wear, and provision must be made for reboring, besides which it must sustain many of the shocks to which the locomotive is subjected while in motion. Hence, while the calculated thicknesses are a trifle less than ⅞ inch it will be found advisable to increase this by 50 per cent or more and make the shells from 1¼ to 1½ inch thick.

The cylinder heads are usually simple in construction and are bolted in the flanges cast at the ends of the cylinder by studs spaced about 5 inches apart from center to center. In the case of the front head these studs have no work to perform other than to withstand the pressure of the steam against the head. At the back they have not only this, but the indeterminate stresses due to the support of the front end of the guides and the varying loads that they transmit to



Figs. 14 and 15. Back and Front Cylinder Heads for Consolidation Freight Locomotive.

the heads. These studs are usually made 1 inch in diameter for the cylinder sizes that are here considered, so that with sixteen studs to withstand the pressure in a counterbore 21¼ inches in diameter, the load on each stud would amount to less than 5,700 pounds per square inch of section of the metal, hence there is an ample margin beyond for safety. The thickness of metal in the heads must be taken independently of any stresses that may be put on them by the steam pressures. The guides must be supported, and there must be provision on the back head for the packing box of the piston rod. Besides this the heads are exposed to external shocks and blows, so that while far in excess of the requirements for resisting the steam pressures, the heads can well be made from 1⅛ to 1½ inch thick. As the heads are especially exposed to the action of the wind, they must be well protected against radiation of heat. At the front, a plain disk head is generally used which is protected by a casing, enclosing an air space in front of the flat surface. Sometimes this space is filled with a non-

conducting material that serves to considerably lessen the radiation. At the back, by the use of a two-bar guide and the flange of the packing box, the construction lends itself very readily to the formation of an air space and pockets for insulating material in the head itself. This is clearly shown in Figs. 12 and 14 of the back heads for the two engines.

Let us briefly review the work to be done on the cylinder and heads. The seat for the smokebox is cast with chipping strips that are chipped to the proper radius, while the two parts (smokebox and saddle) are held together by bolts. The connection between the steam passage and the steam pipe in the smokebox is made by means of a ground joint. The abutting surfaces of the two half-saddles are accurately planed and are held together by bolts that are usually about 1¼ inches in diameter. In common practice these bolts are tapered about 1-16 inch to the foot and they are driven home. The surfaces for the bearing of the frames are planed and the frames and cylinders bolted together with tapered bolts turned to a driving fit. The details of these fastenings will be considered under the subject of the frames. The bore and valve faces of the cylinder are finished with a smooth-cutting tool and are left as they come from the machine. As the joint between the heads and the cylinder ends must be steam-tight, the two surfaces in contact are ground so that no packing is required. Of course in all this work the utmost care must be exercised that a true alignment of every surface is obtained, for without that not only would the working of the moving parts be defective, but the stresses set up would be excessive and the liability of the fixed parts to work loose be greatly increased.

CHAPTER IV.

THROTTLE VALVE—DRY AND STEAM PIPES.

With the four principal items in the construction of the engine decided, namely: the weight, wheel arrangement, boiler, and cylinders, the following steps will be that of the working out of the various details upon which the successful operation of the locomotive depends. Closely connected with the work done in the cylinders and boilers are the means by which the steam generated in the one can be conducted to the other so as to perform its proper functions. In this it is of the utmost importance that the flow of the steam from the boiler to the cylinder should be free and unimpeded and that there should be the minimum drop in pressure when it reaches the steam chest. There must necessarily be some drop, else there would be no flow, but this will decrease as the size of the passages is increased.

The proper sizes of the throttle and the dry pipe are determined, like so many other things in locomotive practice, not so much by a theoretical calculation as by an empirical formula deduced from practice, that has been shown to give satisfactory results. As already stated, the flow of the steam must be free enough to maintain a high pressure in the steam chest, and yet the limits of space in the boiler and smokebox cut down the available dimensions of the pipes to a low figure. In this, too, the designing of a locomotive differs from that of a stationary engine in that the latter is built to run at a constant speed, and with a comparatively uniform degree of admission in the cylinder, so that the steam pipe can be proportioned accordingly and be made to deliver a constant quantity of steam at a uniform rate of flow. In the locomotive, on the other hand, the steam consumption varies between wide limits and is greatest at high speeds when the earliest cut-off is used.

It has been found, then, that the sectional area of the throttle and dry pipe should not be less than one-fifteenth (1-15) that of both cylinders, while the steam pipes in the smokebox should not be less than one-twelfth (1-12) the sectional area of one cylinder. The reason for this apparent discrepancy is that when the engine is working slowly, an area of one-fifteenth can deliver all of the steam required; while, when it is running at a high speed, the point of cut-off is moved back until it never occurs later than at half stroke, so that only one cylinder is taking steam at a time. This makes the dry pipe proportionately larger or raises it to two-fifteenths of the area that it is obliged to supply.

Referring these proportions to the two engines that we have in hand, it will be seen that in a consolidation freight locomotive having wheels 57 inches diameter, and a piston stroke of 26 inches, the maxi-

mum velocity of the piston with the engine running at a speed of ten miles an hour will be about 6 2-3 feet per second, so that at times the flow of steam through the dry pipe would be fifteen times this, or about 100 feet per second. Were the engine to be running at the rate of thirty-five miles an hour the velocity of flow would only be increased to 175 feet per second, because under these circumstances only one cylinder would be taking steam at a time. In the case of the Atlantic passenger locomotive, with 77-inch drivers and a 26-inch

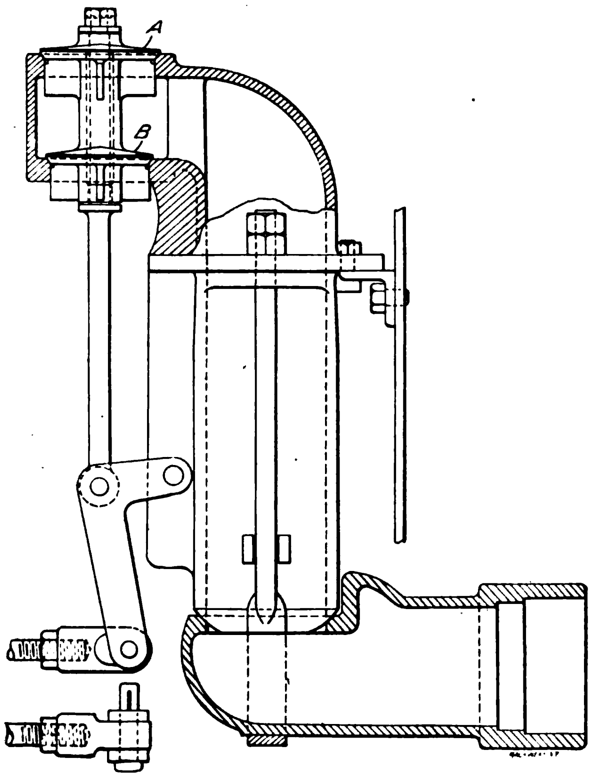


Fig. 16. Section of Ordinary Type of Throttle Valve.

stroke of piston, the velocity of flow through the dry pipe would be, at speeds of fifteen and sixty miles an hour, 111 feet and 223 feet per second respectively.

Such a velocity does not obtain in practice, however, even though the proportions given are maintained, since the steam contained in the steam chest and pipes expands somewhat during admission and is later replenished during expansion and compression; the result is a practical uniformity of velocity of flow through the throttle valve and pipes. Consequently the actual speed is much below that given by the

figures above and does not increase in the direct proportion of the piston speed, as might be expected.

As for the types of throttle valve, dry pipe and steam pipes that are to be used, there is little variation in current practice. The double-seated balanced type of poppet valve is universally used with some slight variations in the details of its construction.

The ordinary throttle valve is shown in section in Fig. 16. In this the two valves, *A* and *B*, close from the top and when raised admit steam to the throttle casting at the top and bottom. In order to

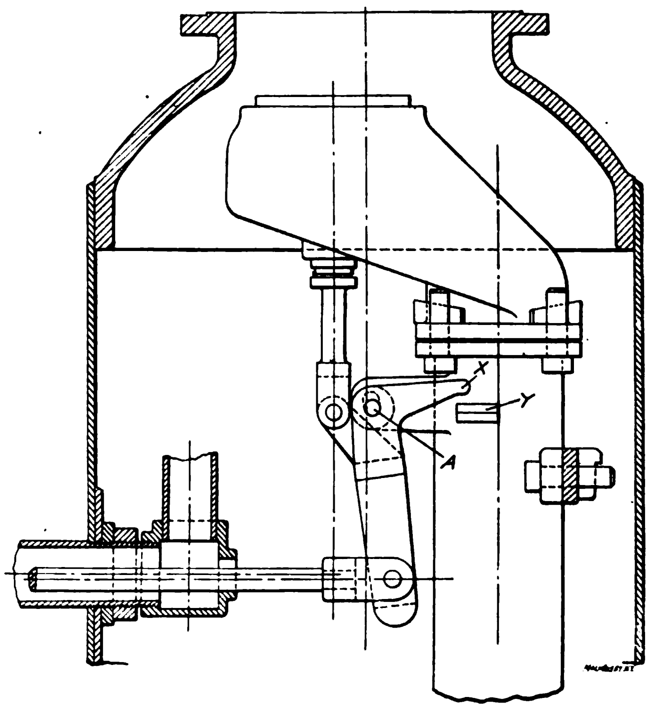


Fig. 17. Section of Throttle Valve with Variable Fulcrum.

assemble this valve it is necessary that the upper should be the larger of the two, so that the lower may be put in from the top. This destroys the perfect balance as the steam, acting upon the greater area of the upper disk, tends to close the valve and will overcome that against the bottom face of the lower disk. This prevents the valve from opening accidentally. The casting may be supported from a bracket bolted to the inside of the dome. As for the inside diameter of this casting, the proportions given above make it a little more than 7½ inches for the consolidation locomotive with 21-inch cylinders, and a little less than 7⅛ inches for the Atlantic type with 19½-inch cylinders. Hence a casting with 7¼ inches diameter of opening can well be adopted for the two. As a matter of fact these dimensions will

be found to vary slightly in practice to meet the exigencies of other requirements that may come up in the working out of the details.

In this connection attention may be called to the fact that the departure from an exact balance of the throttle valve due to the variation in the diameters of the two parts, may cause a considerable preponderance of closing load, especially when high steam pressures are used. This frequently makes it somewhat difficult to open the throttle. In order to avoid this, a system of levers has been designed like that shown in Fig. 17. In this the bell-crank has a double fulcrum. When closed and about to be opened, the bearing is on a pin, *A*. This gives a long leverage and a powerful purchase to assist in the opening of the valve. As the bell-crank turns, the arm *X* comes in contact with the lug, *Y*, after the valve has started and then serves, by the shorter leverage given, to cause the valve to move more rapidly for the same

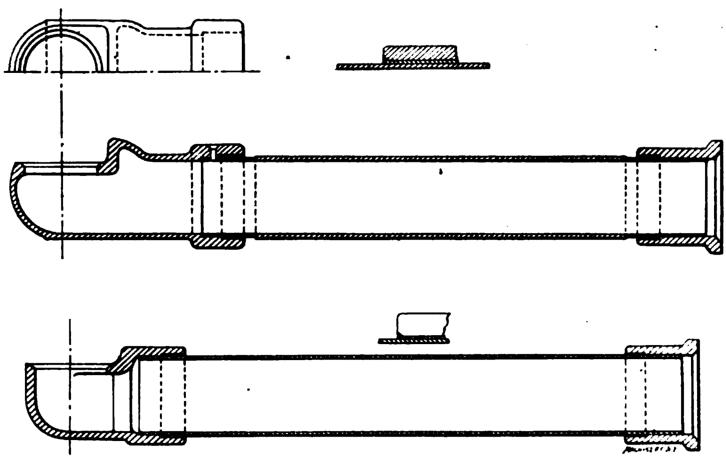


Fig. 18. Dry Pipes for Locomotives.

amount of movement of the throttle stem, the pin *A* rising from its bearing in the slot in the main casting.

The dry pipe is usually made of wrought-iron pipe with cast-iron ends riveted on. The dimensions called for will be the same as those of the throttle casting. But as wrought-iron pipe of these approximate dimensions rises by inches in diameter there is nothing available for the purpose between the nominal diameters of 7 inches and 8 inches. It will, therefore, be necessary to use the former as being nearest to the estimated size, though slightly smaller. A common form of dry pipe is shown in Fig. 18.

It is evident that such a pipe as this cannot be put in through the dome, so it must be run into the hole in the front tube sheet which is, therefore, made large enough to pass the castings riveted to the ends. For a fastening and joint, the arrangement shown in Fig. 19 is used. The dry pipe, *A*, with its castings, is put through the hole in the tube sheet, a heavy reinforcing ring, *E*, having first been fastened against

the inside of the sheet in order to increase its stiffness. The hole in the sheet itself is beveled to match the bevel on the casting. The latter is cut with a spherical ground joint on the outside to take the brass ring, *B*, which is also ground with a flat outer face to bear against the tee-head, *C*. The latter has flanges cast upon either side to which corresponding flanges of the steam pipes, *D*, are bolted. All of these joints are ground and fitted as no soft packing would be able to withstand the intense heat of the smokebox to which they are subjected.

It has already been stated that the area of the steam pipes is put at

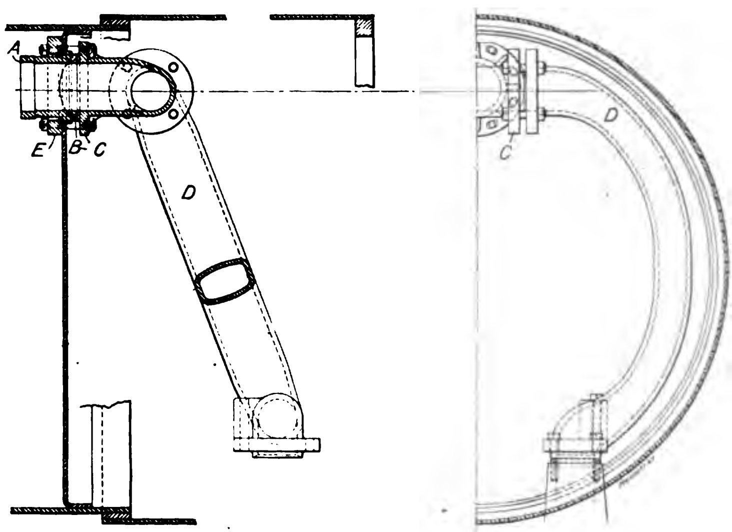


Fig. 19. Tee-head and Steam Pipes.

about one-twelfth that of one cylinder. Hence, for a cylinder 21 inches diameter, the corresponding area of steam pipes would be a little less than 29 square inches and for a 19½-inch cylinder it would be a little less than 25 square inches. Owing to the desirability of encroaching to as slight an extent as possible upon the diametral dimensions of the smokebox, these steam pipes are not made round except at the ends, but are flattened, as shown, so that they lie close to the shell with the longer dimension corresponding with the longitudinal dimensions of the smokebox.

The connection between the foot of the pipe and the cylinder casting is also made by means of a ground joint. The rest of the passage to the steam chest and cylinder has already been treated.

CHAPTER V.

PISTON AND PISTON ROD.

Piston design for locomotive work has been subjected to many changes and there are several forms in use, all of which may be included in two classes: the built-up and the solid. The built-up form of spider bull ring with a follower permits of the removal of the packing without the necessity of removing the piston from the cylinder. Where, however, reduction of weight is of the first importance, a solid piston of a double Z-section, and made of either cast iron or steel casting is used, in which case the packing rings are usually made in one piece and sprung into place. In pistons of this type, when made

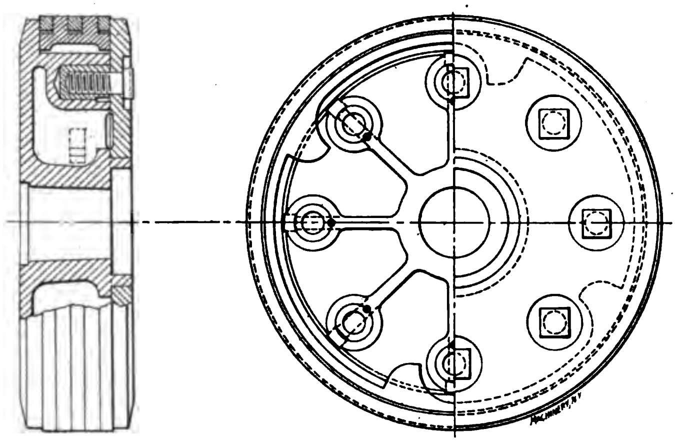


Fig. 20. Piston with Follower and Bull Ring.

of cast steel, the rim is usually surrounded by a cast iron or brass ring, the former being fused or bolted to the body, so as to secure a better wearing surface against the cylinder than the steel would afford. The fused rim is, however, much to be preferred as bolting weakens and adds to the weight of the piston. The first type with the follower and bull ring is shown in Fig. 20, and the second in Fig. 21.

The stresses to which a piston is subjected are of two kinds: one is that of the punching or shearing of the disk about the boss, due to the steam pressure exerted upon it between the rim and the hub; and the other the breaking stress exerted across the diameter. The first should, therefore, be treated as a combination of bend and shear but must be estimated separately. The direct shear alone may be easily provided for by laying out several concentric circles with radii of r, r', r'', etc., and subtracting the area multiplied by the pressure in-

side the circle under consideration from the total pressure on the piston and dividing the remainder by the allowed strength of material per square inch of section. This will give the total area of metal to be used; which, in turn, being divided by the circumference of the circle will leave a quotient equal to the thickness of the metal to be used at the point in question, or the minimum thickness of metal to be used in that particular circle. The first part of the problem may be expressed by the formula:

$$A = \frac{\pi R^2 P - \pi r^2 P}{S} \qquad (11)$$

in which

$A =$ the required area of metal on the given circle,
$R =$ radius of cylinder,
$r =$ radius of section sought,
$P =$ boiler pressure,
$S =$ allowable stress on the material.

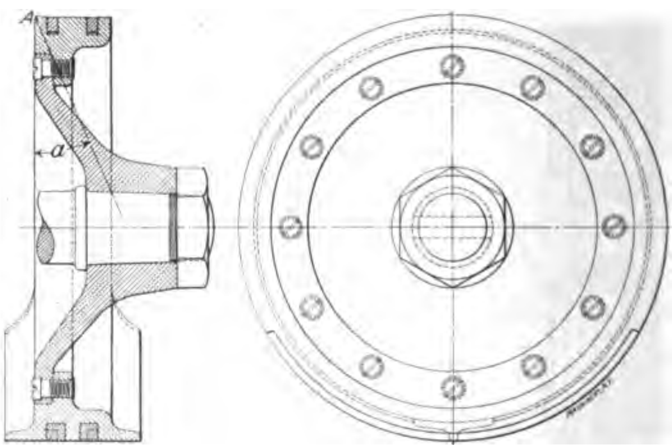


Fig. 21. Cast Steel Z-shaped Piston.

The radial bending stress should be provided for by an ample internal ribbing so as to maintain the requisite stiffness when double-bottomed pistons are used. The disk or single-walled pistons upon which no ribs can be used, should be so designed that the bending stresses are transformed into tension and compression as the loading alternates, and thus transfer them to those sections that would otherwise be subjected to a low stress, such as the rim. This may be accomplished by making the disk conical or of a double Z-section in which a smaller amount of material can be used than in any other form of piston in proportion to the strength developed. It is best, therefore, that the rim should be cast solid with the main body and be made of brass or cast iron so as to secure the full advantage of its wearing qualities.

In providing for the diametral breaking stress across the disk.

which by the way usually provides for the other stresses, as in the one illustrated, the piston may be considered as a beam whose section is that of the piston through its center, with one-half of the total load concentrated on each half at the center of gravity of the same. The distance from the center of gravity to the center of the piston is 0.42 R. Then

$$S = \frac{0.42\,R \times R^2 \times P}{2M} \tag{12}$$

in which

S = working stress of the metal,
R = radius of cylinder,
M = modulus of section.

M must be calculated for each section of piston to which it is desired to apply the formula.

For disk pistons there are a number of empirical formulas in use. One is to take the thickness near the boss as:

$$t = c\,D\,\sqrt{p} \tag{13}$$

D = diameter of piston in inches,
p = pressure per square inch,
t = thickness of metal, and
c = 0.0046 for cast steel,
c = 0.008 for cast iron.

The thickness of the plate near the rim may be taken as 0.6 the thickness at the boss. Calculation now gives for a piston 21 inches diameter a thickness at the boss of about 1⅜ inch and 13-16 inch at the rim for steel; and 2⅜ inches and 1⅜ inch at the boss and rim respectively for cast iron. For the calculation of the thickness of the conical portion of a Z-shaped piston, Unwin gives the following formula for cast steel pistons:

$$t = 0.003\ d\ \sqrt{p} \tag{14}$$

in which

t = thickness of metal,
d = diameter of the cylinder,
p = pressure in pounds per square inch.

For a 21-inch piston working under 200 pounds per square inch of steam pressure the thickness would be 0.88 inch.

After having passed through the various forms of rings expanded by steam and springs, practice has returned to that of a simple split ring held out by its own elasticity. The formulas that are given for the width and thickness of these vary between wide limits, ranging cn a 20-inch piston ring from ⅜ inch to 3 inches. In practice it has been found that on pistons of 18 to 20 inches diameter the width of the rings may be about ½ inch, and for pistons of from 21 inches to 24 inches it may be about ⅝ inch with a thickness ⅛ inch greater than the width; or the width may be expressed approximately as about ⅛ inch less than one-thirtieth the diameter of the cylinder. In a general way it may be stated that the piston packing in ordinary use is that of a snap ring type of from ½ to ⅞ inch square section

to suit the size of the cylinder and of a reasonable tightness. Such rings possess the advantage of simplicity of manufacture and ease of maintenance, and are therefore to be preferred. The only other packing in use is the Dunbar, which still remains a favorite on some roads. It is made in small segments of one L-shaped and one rectangular section, fitted into each other so as to form a ring of square section with the several parts overlapping at the joints. The main objection raised to it is its cost. Great width of packing rings is not required, it being simply necessary to get a good contact around the whole surface, and this can be obtained if the elasticity of the ring is sufficient to give an outward pressure when sprung into position in the cylinder. The size, therefore, is somewhat dependent upon the character of the metal of which the ring is made. It is, of course, desirable to avoid any excess of pressure on account of the wear that would result. The number of rings used varies from two to three with different designers.

The total width of the piston may be placed at approximately one-quarter the diameter of the cylinder, though there are variations from this, the thickness usually being less where but two packing rings are used, when it may be one-fifth.

Piston Rod.

Closely allied with the calculations for the piston are those for the rod, the section of which is, to some extent, governed by the areas through the cotter hole or the bottom of the thread at the smallest part of the rod, where it is attached to the crosshead. As this section of the rod must withstand the full boiler pressure on the piston, it should be of such section that the stress will not exceed 10,000 pounds per square inch of metal. As the piston is usually tapered in its fit in the cross head the main body will be large enough for truing or turning up for wear in the stuffing-box. Still the dimensions should be carefully checked for safety, and the determination of the size of the cotter or key is of the first importance. These cotters are usually subjected to a stress in one direction only as the taper or shoulder on the rod takes that in the other. It is, however, well to calculate this stress as equal to the steam pressure on the whole surface of the piston. The proportions of the cotter must, then, be such as to sustain this full load both in shear and compression. For shear a steel cotter may be safely subjected to 10,000 pounds per square inch of section and to 24,000 pounds in compression. Taken in single shear, the stress will be one-half the total load and the section resisting this will be equal to the thickness multiplied by the width. We thus have the formula:

$$W = 2btS_1 \tag{15}$$

in which

W = load on the cotter,
b = width of cotter,
t = thickness of cotter,
S_1 = shearing stress on metal.

In the case of a 21-inch cylinder working under 200 pounds pressure this formula becomes:

$$69{,}272 = 20{,}000bt,$$

or

$bt = 3.46$ square inches.

If then t is assumed to be ¾ inch, b becomes 4.61 or 4⅝ inches.

The next and most important step in the matter is to determine the diameter of the body of the piston rod. The stresses to which this member is subjected are both in tension and compression; and each must be taken into consideration in calculating the proper diameter. For tensile stress we have

$$d = 2\sqrt{\frac{W}{\pi S}} \tag{16}$$

in which

d = diameter of the rod,

W = total load on the piston,

S = allowable working stress on the material = 10,000 pounds per square inch.

By substitution, we have, for a 21-inch piston

$$d = 2\sqrt{\frac{69{,}272}{31{,}416}} = 3 \text{ inches nearly.}$$

The determination of the compressive strength is a much more complicated matter owing to the fact that the piston rod cannot be considered as a strut pure and simple, because of its motion and because of the bending stresses that are put upon it by the looseness of the crosshead in the guides and the piston in the cylinder. A number of empirical formulas have been proposed for this work, but none of them has as yet been accepted as adapted to all conditions on account of the variation in the results that have been obtained in experiments conducted upon a large scale. It is generally considered, however, that when the length of a column is less than twelve times its diameter, the compression can be safely placed at the same figure as the tension, and this is the case with nearly all locomotive piston rods. Cases, however, may arise when the compression would be excessive and, in such, it is well to refer to Rankine's formula for columns with round, free ends, which is

$$\frac{P}{A} = \frac{S}{1 + \dfrac{q\,l^2}{r^2}} \tag{17}$$

in which

P = the total load,

A = area of the section of the rod,

S = ultimate compressive strength of the material = 150,000 pounds per square inch,

l = length of rod in inches,

r = radius of gyration $= \dfrac{d}{4}$,

$$q = \frac{4}{25{,}000} \text{ for steel,}$$

$$q = \frac{4}{36{,}000} \text{ for wrought iron.}$$

This formula can be readily changed to terms indicating the diameter, by substitution and a proper allowance of a factor of safety for S.

A similar formula of Merriman is

$$C = \frac{B}{1 - \frac{n\,B}{10\,E} \times \frac{l^2}{r^2}} \qquad (18)$$

in which

C = the maximum compression stress per square inch of area,

B = the load per square inch of section of the rod,

E = modulus of elasticity = 30,000,000 for steel; 25,000,000 for wrought iron,

n = 1 for round end bearings and ¼ for square ends.

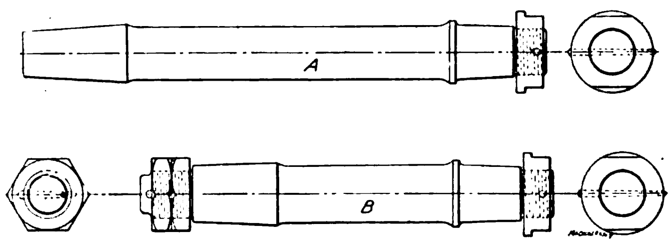


Fig. 22. Piston Rods for Locomotives.

For an approximate estimate, we may use the formula:

$$W = \frac{1125\,\pi\,d^4\,S}{4500\,d^2 + 4\,l^2} \qquad (19)$$

in which

l = the length of the rod in inches,

S = allowable stress per square inch of section.

For the determination of l it will be necessary to lay out the rod on the drawing board so as to allow for the necessary clearances, and the length, in the case of the consolidation cylinder with 26 inches stroke, will be found to be about 38 inches from the back face of the piston to the boss on the crosshead.

Then assuming the maximum fiber stress to be 9,000 pounds per square inch the equation (19) becomes

$$69{,}272 = \frac{1125 \times 3.1416 \times d^4 \times 9{,}000}{4500\,d^2 + 4 \times 1444}, \text{ from which } d = 3.3.$$

As some allowance must be made for the truing up of the rod on account of wear the addition to this amount will depend upon the personal opinion of the designer, modified, to an extent, by the character

of the road upon which it is to work, a greater addition being made for a sandy track than for one that is rock ballasted, because of the greater wear likely to take place on the former. Taking ⅜ inch as a proper allowance for this purpose, the rod becomes 3 11-16 inches in diameter and had therefore best be made 3¾ inches to avoid odd measurements.

Fig. 22 gives the two forms of piston rods in most common use in the United States. In the upper elevation, *A*, is shown the form used when it is attached to the crosshead by a cotter; the lower, *B*, shows one fastened by nuts. It may be added in conclusion that piston rods and keys should be made of a high grade steel whose tensile strength does not fall below 65,000 pounds per square inch.

It now remains to be seen whether there will be metal enough left in the rod, after cutting out a keyway ¾ inch wide, to allow sufficient strength to carry the load imposed. This may be expressed by the formula:

$$W = \left(\frac{\pi d^2}{4} - d\,t\right) \times S \qquad (20)$$

in which

d = diameter of piston rod,

t = width of keyway,

S = allowable tensile stress to be put on the metal.

Assuming $S = 10{,}000$, the formula becomes

$$W = (11.04 - 2.81) \times 10{,}000 = 82{,}300,$$

a result far in excess of the load of 69,272 pounds that it will be required to sustain; and this margin is still further increased by the enlargement of the ends as shown in Fig. 22.

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LOCOMOTIVE DESIGN

By Geo. L. Fowler and Carl J. Mellin

PART II.

VALVE MOTION

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VALVE MOTION

CONTENTS

CHAPTER I.

THEORY OF VALVE MOTION.*

Next in importance to the boiler in determining the efficiency of the locomotive as a whole, is the valve motion, and too much stress cannot be put upon the value of a proper design for this element in the machine. The Stephenson link motion, which may be said to be the one universally used upon American locomotives, possesses the peculiarity of being exceedingly sensitive to a clcse adjustment of all of its parts in order that a correct action and proper distribution of the steam may be obtained; while with the roughest kind of haphazard design or no design at all, it will do its work after a fashion and make the wheels go round. It is evident, however, that in order that the steam distribution in the cylinder may be as efficient as possible for all speeds and all points of cut-off, the utmost care must be exercised in the designing of the valves and machinery by which they are driven.

Up to within a few years the flat slide valve was in universal use. At first it was the common unbalanced D-valve, which was followed, as the steam pressure was increased, by the balanced valve, while the current practice, where high pressures are used, lies between the flat balanced valve and the piston type. The piston valve possesses some advantages over the flat valve in that it is fully balanced, though the slide valve can be quite satisfactorily designed in this respect. In short, it is a matter of choice and convenience of construction and maintenance as to which shall be used, though the tendency of modern practice is toward an increasing use of the piston valve.

The main difference to be considered in the designing of the two types of valves is that while the flat slide valve is invariably arranged for an outside admission of steam to the cylinder, the piston valve should be arranged for an inside admission. This is not absolutely necessary, but the advantages are that the steam passages can thus be better protected from the cold and radiation, and the steam chest heads and packings relieved from all pressure except that of the exhaust steam, which is but a few pounds above that cf the atmosphere and so puts really very little stress upon these parts. If the piston

*The present number of MACHINERY'S Reference Series is the second part of a treatise on complete Locomotive Design, covered by Nos. 27, 28, 29 and 30 of the Series, and originally published in RAILWAY MACHINERY (the railway edition of MACHINERY). Each of the four parts of the complete work treats separately on one or more special features of locomotive design; and while the four parts make one homogeneous treatise on the whole subject, each part is complete by itself. In order to give concrete form to the examples and theoretical considerations, it is assumed that a consolidation freight locomotive and an Atlantic type passenger engine are being designed. It is further assumed that these locomotives are designed for a division 150 miles long, laid with rails weighing 75 pounds per yard, and with a ruling grade of one per cent ten miles in length.

valve is so designed that the admission is on the inside, it should be made hollow with as large a passage through the center as possible. The object of this is to secure a large area for the movement of the exhaust, so that, at the instant of release, the pressure may be reduced to a minimum. The use of the hollow valve facilitates this by permitting an escape through the exhaust passages at each end; and these latter should be carried to the base of the exhaust pipe and made to meet at a very acute angle so that the two currents will combine without forming obstructive eddies and thus aid in the fanning of the fire. Such a combination will be found to permit of the use of a larger exhaust nozzle than is possible with a solid valve.

As for the mechanisms by which the valve is moved, there are four, or as they are sometimes classified, five different systems that are in use upon locomotives, and which are named after their designers, *viz.:* the Stephenson, Gooch, Allan, Walschaerts, and Joy.

The first is known as the shifting link motion, and is that mostly in use in the United States. The second is directly opposite in its

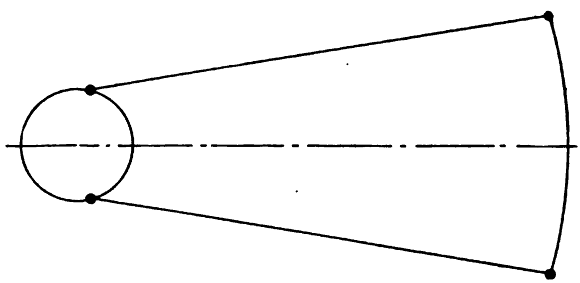

Fig. 1. Open Eccentric Rods.

action, in that the link is stationary and the link-block, attached to the valve-rod, is moved up and down. The Allan is a combination of the first two in that both the link itself and the valve-rod are shifted, by which it becomes possible to make the link straight. This, of course, greatly simplifies the construction and maintenance of the link, and the motion is extensively used in Europe. All of these motions are operated with two eccentrics, one for the forward and the other for the backward motion.

In the Stephenson and Allan motion when the eccentric-rods are open, as in Fig. 1, the lead is increased as the link is hooked up and the point of cut-off made earlier. If, however, the rods are crossed as in Fig. 2, the hooking-up of the link reduces the lead, though this reduction is much less than the increase in the former case. Again, with either open or crossed rods, the corresponding increase or reduction of lead is much less with the Allan than with the Stephenson valve motion. With the Gooch, Walschaerts, and Joy motions the lead is constant for all points of cut-off.

It will be seen, by reference to the diagrams Figs. 1 and 2, that the eccentric-rods are said to be "open" when, with the eccentric centers

upon the same side of the axle as the link, the rods are not crossed; and "crossed" when the rods do cross in that position. This holds true whether the motion be transmitted direct to the valve stem or indirectly through a rocker arm. This term of direct or indirect application refers to the relative movement of the valve and the link-block. When they move in the same direction the motion is said to be "direct"; when in the opposite directions, "indirect."

The Walschaerts gear is driven by a combination of an eccentric or short-stroke return crank from the main crank-pin and a connection to the crosshead. The Joy gear is driven from a connection to the connecting-rod. The former has been extensively applied on the continent of Europe and the latter to some extent in England. The Walschaerts gear has recently been applied to several of the largest engines built in the United States with such pronounced success that the use is being extended, and some engineers have expressed the opinion that the prospects are that it will eventually take the lead over

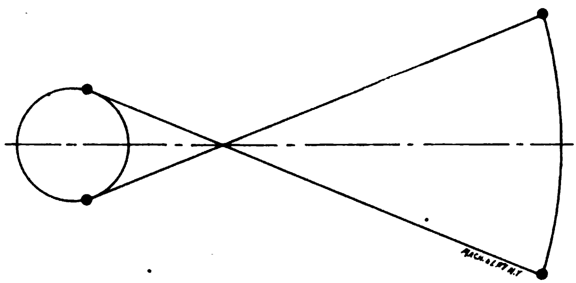

Fig. 2. Crossed Eccentric Rods.

the Stephenson gear, as it has abroad, on account of the many advantages which it possesses and which have not been properly demonstrated in American practice until lately.

As the Stephenson motion is the one mostly used in this country it will be first considered. It is probably the most flexible of any in use, and can be most readily adapted to irregularities in the running and operation of the machine. At the same time it will get out of adjustment very easily and requires the utmost care in its designing in order that it may work properly. With this as an introduction we may now enter upon an examination of the motion and its requirements. To work out the valve motion theoretically and mathematically is a long and tedious operation. As its application and development for use upon a locomotive involves a number of irregularities that are necessarily neglected, when treating the subject from a purely mathematical standpoint, this method will be simplified by use of diagrams by which the steps needed for its development for actual use will be shown.

In the first place, the introduction of a rocker-arm between the eccentrics and the valve serves as a means of increasing or reducing the travel of the latter as compared with the throw of the eccentrics

as well as transferring the line of motion to suit convenience in locating valves and valve spindles. The angularity of the connecting- and eccentric-rods introduces irregularities that can, to a great extent, be compensated by the location of the link-saddle pin. The longer these rods can be made the better for the action of the motion, and it is well not to make them too short, when it would become impossible to compensate for the irregularities that would be introduced.

Experience and calculations have shown that, to secure a satisfactory action of the valve motion, the connecting-rod should not be less than six times the radius of the crank, that is, three times the piston stroke, and that the eccentric-rods should not be less than eight times the throw of the eccentric. As a matter of fact, the eccentric-rods are usually of a greater length than this. In the case of the link the radius should be equal to the distance from the center of the link-block, when in its central position, to the center of the axle, and the distance between the eccentric-rod pins should not be less than two and a half times the throw of the eccentric. If it is less than this the angle assumed by the link relatively to the block will be such that the slip of the latter will not only be excessive, but it will be apt to stick and put undue stresses on the entire mechanism of the motion.

As already intimated, the irregularities introduced by the angularity of the rods may be compensated for by adjustment of the location of the saddle-pin, which will always be inside the center line of the arc. If, however, it is carried too far, it will give an objectionably long slip to the link, especially if the rod lengths are near their minimum. So, as this offset of the saddle-pin, as it is called, is less with an outside admission valve and an indirect motion, or, what is the same thing, an inside admission and direct motion, than where the contrary conditions exist, it is always best to use one or the other of these two combinations when no other advantage is to be gained by a reversal of the conditions. The adjustment of the valve also requires that particular attention should be paid to the lead that is obtained at full travel, as well as the increase resulting from a linking-up of the engine, so that, when running with an early cut-off the lead and pre-admission may not be excessive.

This can best be studied by a consideration of the effect of the movement of the link on the travel of the valve. With the link in full gear the block is usually so related to the pin of the corresponding eccentric-rod that its motion to and fro is equal to the diameter of the path of the controlling eccentric. Then, if the two rocker-arms are of the same length, the travel of the valve will also be the same. In other words one eccentric controls the valve and the other merely causes the link to oscillate about the block, as far as relative motions are concerned. But, when the link is raised, both eccentrics have an effect on the motion of the valve, and the resultant of this is as though another and controlling eccentric of a shorter throw were to be introduced between the two. The throw of this resultant eccentric decreases as the link is raised until mid-gear is reached, when the throw is at the minimum. Meanwhile its position has shifted from the cen-

ter of the forward eccentric to a point in line with the crank and midway between the two actual eccentrics. At this point the radius is equal to the sum of the lap and lead in mid-gear. As the link is raised still further, the radius gradually increases and shifts its center until, in full gear back, it coincides with the center of the backing eccentric.

The center of this imaginary or resultant eccentric has then traversed a path that is a parabolic curve connecting the centers of the two eccentrics and passing through a point in line with the crank and distant from the center of the axle by an amount equal to the sum of the lap and lead in mid-gear. The height of this curve or its distance from a straight line connecting the two eccentric centers is equal to the increase of lead between full and mid-gear. For a given maximum cut-off and valve travel, the required angular advance

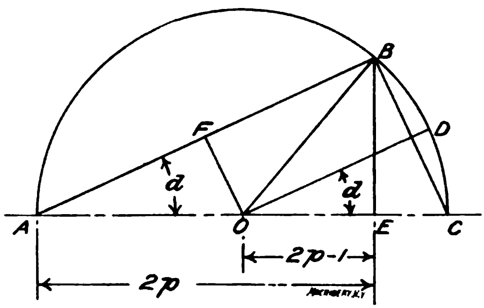

Fig. 3. Diagram for Finding Angular Advance of Eccentric.

and lap can be calculated, when no lead is allowed in full gear or an excessive pre-admission is to be avoided, by the following formula, based on a radius equal to 1:

$$\sin d = \sqrt{1 - p} \qquad (1)$$

where $\sin d =$ lap, or lap and lead,
$p =$ cut-off in hundredths of the stroke, and
$d =$ angular advance of eccentrics.

If r equals radius of the throw of the eccentrics, then multiplying the second term of the equation by r the formula gives the value of $\sin d$ as follows:

$$\sin d = r\sqrt{1 - p} \qquad (2)$$

An analysis of this formula is readily made by reference to diagram Fig. 3. Draw the semicircle ABC to represent the path of the center of the eccentric, with a center at O and a radius equal to 1. Lay off the distance $AE = 2p$ and draw EB at right angles to the diameter AOC. Connect the points A and B and B and C, and from O draw OF at right angles to and bisecting the line AB, and also draw OD parallel to AB. With this construction the arc AB will be equal to that swept through by the crank and the eccentric from the beginning of the

stroke to the point of cut-off, and the arc DC which is one-half of BC will be equal to the angular advance of the eccentric. Then the angle DOC equals d.

From the diagram

$$AC : BC = BC : EC.$$

By multiplying and substituting the value $AC = 2$ and $EC = 2 - 2p$ we have

$$BC = \sqrt{2(2-2p)} = 2\sqrt{1-p}$$

and

$$\frac{BC}{2} = \sin d = \sqrt{1-p}$$

For example, suppose a maximum cut-off of 0.85 of the stroke is desired with no lead in full gear and a 6-inch throw of the eccentric

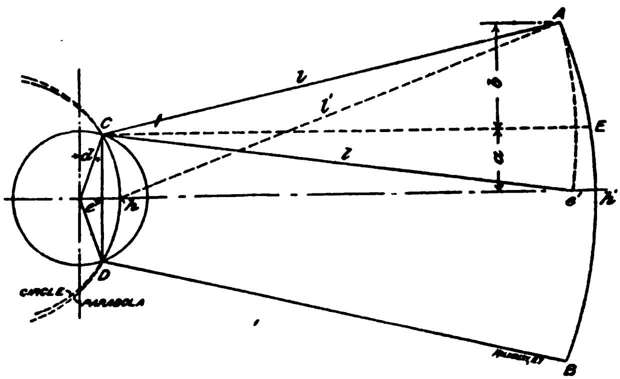

Fig. 4. Link Motion Diagram Illustrating Increase of Lead with Open Rods.

and travel of valve. The advance of the eccentric and the lap of the valve will be obtained by substituting these values in the formula:

$$\sin d = 3\sqrt{1-p} = 1.16 \text{ inch,}$$

or nearly 1 3-16 inch lap without lead. The radius being 3 inches the value obtained based on the radius as a unit must therefore be multiplied by this amount, as above, in order to get the result expressed in inches as provided for in formula (2) and in the example given.

Having thus determined the angular advance and lap of the eccentric it remains to obtain the positions of the virtual eccentric as the engine is hooked up towards mid-gear. It is now in order to examine the curve that the center of the virtual or resultant eccentric traverses in passing from the point of full gear forward to full gear back.

Let the diagram, Fig. 4, represent the eccentrics and the link, in which the circle CD is the path followed by the actual centers about the axle. C will then be the point occupied by the forward eccentric when the crank is on the center, and D that of the backing eccentric. The first step in the problem is to determine the height eh or distance of the resultant eccentric center from the chord CD, when the link is in mid-gear. This height eh is equal to the displacement of the

valve caused by moving the link from full to mid-gear. In Fig. 4

d = the angular advance of the eccentric,

$a = \dfrac{CD}{2}$,

b = one-half the distance between the pivot points of the link, minus a,

$2(a+b)$ = distance between pivot points of the link,

l = length of eccentric rods,

l' = radius of the link.

When the link is in full gear forward the eccentric-rod occupies the position Ce', and when the link is in mid-gear the rod occupies the position CA. It will be seen that, in moving from e' to A, the pivot point of the link moves away from the vertical center line, passing through the center of the axle until it reaches the line CE and after that approaches the same until it reaches A. The amount of separation may be expressed as equal to

$$l - \sqrt{l^2 - a^2}$$

and the approach as

$$l - \sqrt{l^2 - b^2}$$

which would make the total approach

$$[l - \sqrt{l^2 - b^2}] - [l - \sqrt{l^2 - a^2}] = \sqrt{l^2 - a^2} - \sqrt{l^2 - b^2}$$

At the same time the center line of the link that is occupied by the link block has moved to the point h', which is that of the center when the pivot points are at A and B. This distance, $e'h'$, through which the block is moved, is equal to the versed sine of the angle Ahh', less the distance by which the pivot point A actually approached the vertical passing through the center of the axle. If the radius of the link be taken as equal to the length of the eccentric-rods, the center of the link in mid-gear will be at the point h.

The versed sine of the angle Ahh' will then be

$$l' - \sqrt{l'^2 - (a+b)^2}$$

Hence

$$e'h' = l' - \sqrt{l'^2 - (a+b)^2} - (\sqrt{l^2 - a^2} - \sqrt{l^2 - b^2})$$

Then

$$hE = hh' = Ce' = l = l'$$
$$\text{and } eh = e'h'$$

In this case the radius of the link l' is assumed to be equal to the working length of the eccentric-rod l. An inequality in these dimensions makes no difference in the accuracy of the formula.

According to Prof. Zeuner in his analysis of the Stephenson valve motion, the curve passing through the points ChD is a parabola, but between the limits C and D in which it is used, it coincides so closely with a circle that it may be regarded as one whose radius is expressed by the formula:

$$r = \frac{B^2 + a^2}{2B} \qquad (8)$$

in which

$$B = eh.$$

Before taking up the working out of the details of the valve motion mechanism, a few diagrams will be presented illustrative of the various conditions that will be encountered in practice.

Fig. 5 is a diagram showing the action in full gear conditions. To construct it, draw the circle *CDGH* with the diameter equal to the travel of the valve. From the same center and with *N* equal to the lap of the valve, draw the lap circle. At the extremity of the horizontal diameter with the lead in full gear as a radius, draw the small lead circle *C*. Draw the line *EF* tangent to the lead and lap circles; then parallel to it the dotted line *CD* which defines the point *D;* and also the line *GH*, defining these two points as well. By drawing *FP* at right angles to *AB*, the point of cut-off *P* is located and can be measured from *C*, in the percentage of the diameter of the valve travel circle to which it will bear the same relation as the piston position, at that instant, bears to the stroke of the engine.

If the point of maximum cut-off has been decided upon, the process may be reversed by laying off the desired cut-off point *P* on the line *AB* and drawing line *PF* to intersect the travel circle. Draw the lead circle as before, about *C*, and then lay down the line *EF* tangent to the same. The lap is the perpendicular distance from the line *EF* to the center *O* and its circle may be drawn tangent to *EF*. The line *GH* is drawn as before parallel to *EF*.

These few lines practically define all of the points that will be needed for a study of the valve action at full gear when the exhaust lap is zero or line and line. The line *OE* indicates the crank position when the valve opens and *OF* when it closes at the cut-off. The exhaust opens at *G* and compression begins at *H*, and by projecting these points to the diagram below, it is possible to obtain the prominent points cf an indicator diagram for full gear action.

When the center of the axle lies in the axis of the cylinder and valve motion the crank will coincide with the line *KL* at the instant that the centers of the two eccentrics are at *C* and *D* respectively. The curved line connecting *C* and *D* indicates the locus of the virtual or resultant eccentric, that operates the valve under the combined influence of the two eccentrics, and its distance from *EF* at any point represents the lead that the engine is working with, when hooked up so that the center of the virtual eccentric coincides with the given point.

Fig. 6 is a diagram showing the effect of raising the link and thus shortening the period of steam admission. In a general way it closely resembles that given in Fig. 5. A change has, however, been made by the introduction of a small exhaust lap, in order that the effect of delaying the exhaust opening at *G* and advancing the compression at *H* may be seen. By raising the link so that the virtual center of the eccentric is at *T*, the valve travel becomes equal to the diameter of a circle drawn through *T* with *O* as a center. The lead at this

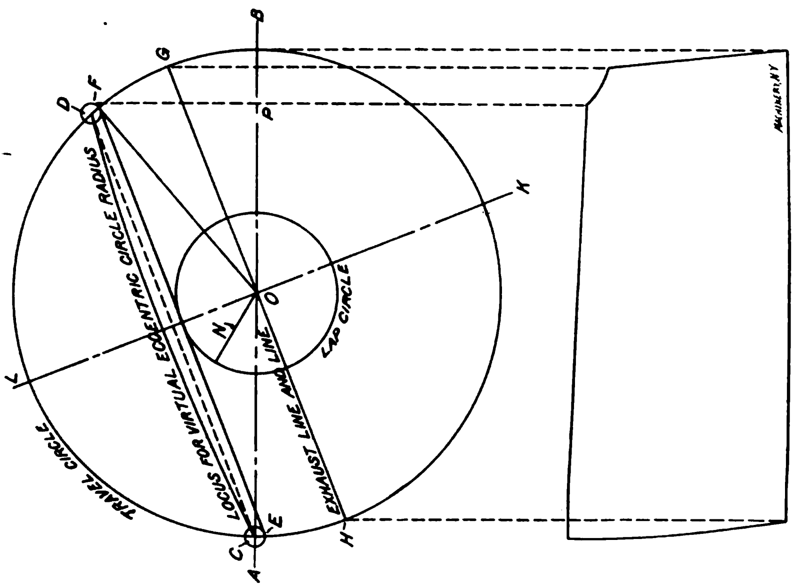

Fig 5. Diagram showing Conditions when in Full Gear.

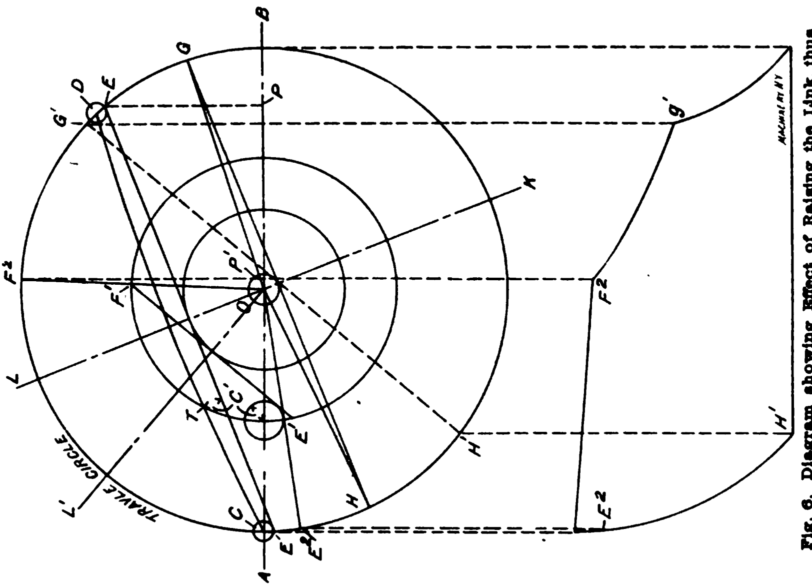

Fig. 6. Diagram showing Effect of Raising the Link, thus shortening the Period of Steam Admission.

point, being greater than that used for the describing of the circles at C and D, is taken as the radius for drawing a circle at C' where the circle passing through T intersects the horizontal diameter. The line $E'F'$ is then drawn tangent to this new lead circle as well as to the lap circle, intersecting the new valve travel circle at F''. The radius OF'' is then drawn through this point (F'), which is, in turn, projected upon the line AB which it intersects at P'. This latter point indicates the point of cut-off or valve closure which, in this instance, is at about half-stroke.

With the change in the angle between the center line AB and the cut-off line $E'F'$, there is a corresponding change in the exhaust and compression line, the two remaining parallel, so that the point of exhaust opening G is advanced to G' and that the commencement of compression H is advanced to H', while preadmission is moved ahead to E''. It thus appears that, with the advancement of the point of

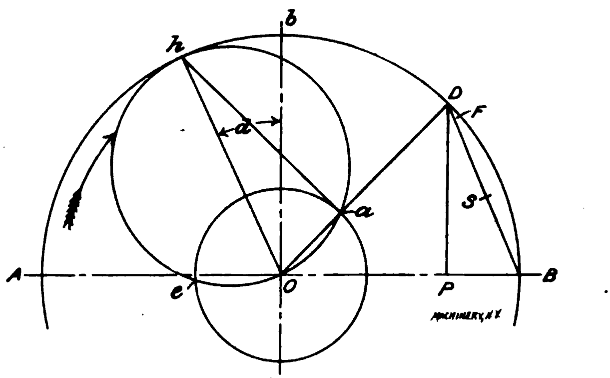

Fig. 7. Zeuner's Diagram.

cut-off, the same thing is done with the opening and closing of the exhaust and the lead opening. If these points are then projected upon the line AB. their location with reference to the stroke of the piston is at once fixed; while by extending them below, the outline of an indicator card is obtained.

Fig. 7 is the well-known Zeuner diagram. It is a method of graphically determining the valve events and was devised by Prof. Zeuner, of the Polytechnic Institute of Zürich, Switzerland, and is, therefore, known by his name. It offers a simple and reliable means of ascertaining not only the main valve events, but the amount of port opening both for steam admission and exhaust, for any piston or crank position throughout the entire stroke, and for any desired link position of the gear.

The angle of advance is obtained by formula (2) when the lap in full gear is equal to sin d or, in other words, when the valve is set with no lead, or by this diagram, where the lead can be introduced without much complication. It will be noted, however, that the angle

of advance d must be laid off towards the crank, or 90 degrees — d in advance of the same, in order to obtain a correct reading.

In the construction of the diagram, let AOB be the diameter of a circle upon which the semi-circle ADB is described. The desired point of cut-off is laid down at P, the period of admission extending from A to P. From P a line PD is drawn at right angles to AOB to D, which is then connected with B by the line DB. On this line DB lay off DF from D equal to the amount of lead that is desired; and divide BF into two equal parts at S. Then $BS = FS$ will be the required lap.

With BS as a radius draw a circle about O, as a lap circle, and from the point a where it intersects the line OD draw ah at right angles

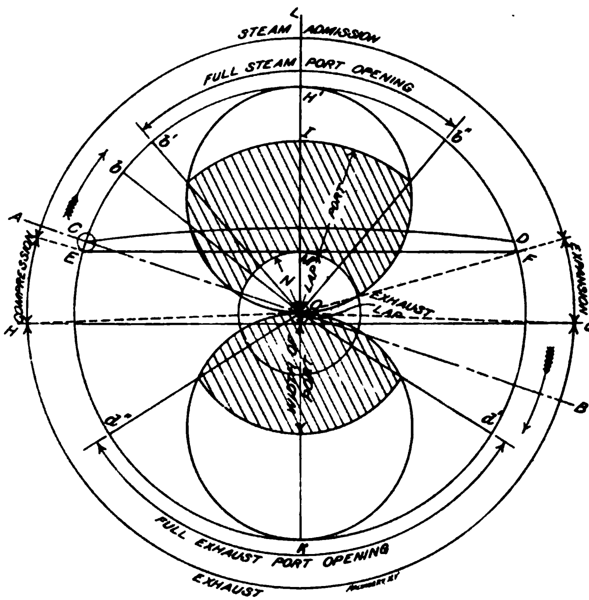

Fig. 8. Combination Zeuner's Diagram.

to OD. Draw the radius Oh and with this as a diameter draw the valve circle $Oeha$, and the angle of advance will be found to be equal to hOb. The application of this diagram will be set forth in Figs. 8 and 9.

In Fig. 8, which is a combination diagram, the lap and lead circles as well as the positions C and D of the eccentrics are the same as in Fig. 5, when the crank is at K for a direct valve motion; at L for an indirect. A modification has, however, been introduced in the shape of a small exhaust lap for the purpose of illustrating its appearance on the diagram. On this diagram, as thus constructed, the Zeuner diagram is imposed, by means of which the port opening at any point of the stroke, or rather at any crank angle, can be determined under both the admission and exhaust periods.

By locating this diagram relatively to the crank, the center line of

the diagram will make an angle equal to that of the angle of advance, but on the opposite side of the 90 degrees to that of the actual location of the eccentric, or at a point where the eccentrics would be when the crank had reached *B*, and was turning in the opposite direction. With this location of the diagram it will be found that its centers will fall on the line *OL* of the original diagram, Fig. 5, which is one at right angles to *EF*. As the diameter of the valve circle is equal to the radius of the eccentric circle, it will be seen that the former intersects the lap circle at the same points as *OE* and *OF* by which the opening and closing of the valve is indicated respectively.

On the line *OL* lay off the distance *IM* outside the lap circle equal to the width of the port and draw the port circle through *I* with *O* as a center. By cross hatching that portion of the area of the valve circle lying between the lap and port circles a graphical representation of the port opening is obtained. It is now possible to find the actual port opening for any position of the crank throughout the admission period, wherever a radius is drawn. For example, when the crank is on the line *Ob*, the opening will be about three-quarters the full width of the port, whereas at any point between *b′* and *b″* there is a full width of port opening. At the latter point the valve commences to close, an act that is finally accomplished when the crank reaches *OF*.

The exhaust valve circle is located on the opposite side of the center and is of the same diameter as the steam circle. With the small exhaust lap as a radius, a circle is drawn about *O* as a center, from which the width of the steam port is laid out along *OK* and the port circle drawn just as on the steam side. The lines *OG* and *OH* indicate the exhaust opening and commencement of compression respectively, while the lines *Od′* and *Od″* indicate the limits of the crank positions where there is a full port opening. A study of this combination diagram will show that all of the valve events coincide exactly in the two that have been thus superimposed, throughout the entire revolution of the crank.

Fig. 9 is a Zeuner diagram laid out to show its adaptability to the determination of the several valve events for the different points of cut-off. From the diagrams already discussed the full gear location of the eccentric was found, which is here indicated as r^5. It also appeared in the discussion of the diagram given in Fig. 6, that the line *OL* representing the center line of motion moves more and more towards the center line of the crank as the link is raised and a shortened cut-off obtained; until, in mid-gear, the two coincide. And, further, that the intersection of these lines and the circles forms the radius of the virtual eccentric and indicates the positions of the latter when the crank is at *A* or the beginning of the stroke..

At the various points of cut-off indicated by r', r^2, r^3, etc., the center of the virtual or resultant eccentric at R', R^2, R^3, etc., falls on the line *CD*, which represents the locus of these radii, and the diameters of the circles passing through these points are equal to the travel of the valve at the corresponding link positions. For the sake of simplicity no exhaust lap or clearance is shown in the diagram, Fig. 9,

but the valve edge is considered to be line and line. Under these conditions the exhaust and compression points fall at right angles to the radial line of these several positions. Thus, the exhaust opening of the position corresponding to R^3 is at e^3, while its closure and the commencement of compression occurs at f^3. The same, of course, holds true of the other positions. If R^3 is located, then r^3, or the point in the revolution of the crank where the valve closes is found by means

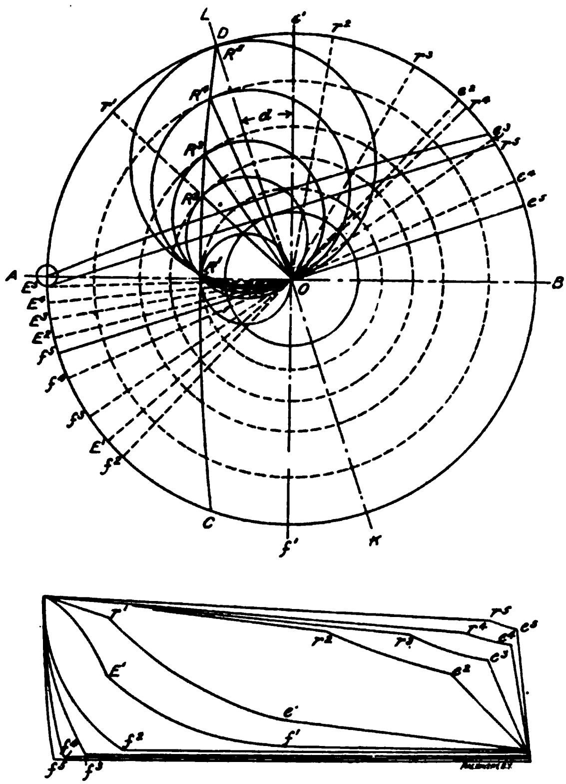

Fig. 9. Zeuner's Diagram, Illustrating the Change of Valve Events for different Points of Cut-off.

of a radius drawn through the point of intersection of the lap and valve circles, the latter having the line OR^3 as a diameter.

If, on the other hand, the location of R^3 or the center of the virtual eccentric is to be obtained from the predetermined point of valve closure as at r^3, the radius Or^3 is drawn; and, through the center O and the point of the intersection of the radius with the lap circle, a circle is drawn with a diameter of such a length that, when a circle is drawn about O as a center with this diameter as a radius, the larger and smaller circles shall be tangent to each other at the point where they intersect the locus CD of the virtual eccentric. The method

was fully explained in connection with Fig. 7. An examination of the diagram will show this to be the case with all of the points of intersection R', R^2, R^3, R^4 and R^5 that have been laid down.

The several points of preadmission E', E^2, E^3, etc., are located in the same manner by drawing radii through the other intersections of the valve and lap circles. By projecting the several events it becomes possible to construct the outlines of the indicator diagrams that are shown at the lower part of the figure, thus facilitating the study of the action of the valve by the clearer detail so obtained.

Thus far attention has been directed solely to the forward motion. In the consideration of the back gear, it will, of course, be found that exactly the same conditions will obtain, but they will appear in a reversed position on the diagram. That is to say, the valve circles should be drawn below the line AB while the exhaust can be obtained by projecting each eccentric position line to the opposite side of O.

With the principles as set forth in these diagrams well in mind, it becomes possible to make an analysis of the consecutive events of the motion of the valve at various rates of admission and to enter intelligently upon the task of working out the details of the mechanism by which the valve is to be operated, and the theoretical studies can thus be put into a tangible and practical form.

CHAPTER II.

CALCULATION OF VALVE DETAILS.

With the study of the movement of the valve and the eccentric thoroughly in mind, the next step is to so proportion the various parts of the gear that an efficient distribution of the steam will be obtained. It must be borne in mind, however, in this work, that many of the ratios and proportions that will be given are outlines only, and that variations from the figures are allowable and even required in order to meet the exigencies of design and construction.

As an example of this, in starting at the cylinder, the ports in the valve face should be made with an area of from 1/10 to 1/12 that of the cross sectional area of the cylinder. This is, in itself, quite a range, but is necessary on account of the great differences in the amount of steam that has to be admitted to the cylinder by the different classes of engines, such as fast express, road freight, and switching. With 1/10 or 1/12 the cylinder area as the port area, the next step is the proportioning of the length of the latter to the width. In this the width should be, or rather may well be, made about 1/12 the length, subject to necessary variations. And finally the average travel of the valve may be put at about three and one-half times the port opening. These are the rough starting figures, and the lap of the valve may be determined either from the formula (2) or fixed arbitrarily at from one-fifth to one-sixth the travel of the valve in full gear. The use of the formula is to be preferred as the latter method leaves much uncertainty as to what the point of maximum cut-off will be and may involve irregularities in the equalization of the same on the two strokes.

In the days when 16 inches represented the standard or maximum diameter of cylinder, the valves were small and light, and the pressure of steam upon their backs did not cause enough frictional resistance to necessitate balancing. But, with the increase in cylinder diameters and steam pressures, the larger valves that are used call for a balancing so as to relieve the rods of the tremendous stress that would otherwise be imposed upon them in the work that they are called upon to do. This balancing may be accomplished by any one of the accepted methods.

It should remove the steam pressure from the back of the valve over an area amounting to the sum of the areas of one steam port, the exhaust port and the two bridges, plus 8 per cent of this sum for plain valves and plus 5 per cent for Allan-ported valves. The reason for the use of a smaller amount with the Allan-ported valve is that steam cuts under and through the port and balances its own area when one of the openings is covered by the seat.

This balancing is introduced in order to lessen the resistance of the valve under ordinary working conditions; but, in calculating the dimensions of stems and rods, it is necessary to consider the work that would have to be performed in case the balancing strips were broken and the full pressure were to be put upon the back of the valve. The frictional resistance of the valve on its face may be taken as 20 per cent of the total load or 20 per cent of the valve area multiplied by the boiler pressure. Of course such a high coefficient of friction would only obtain if the valve face were very dry, but this is exactly what has to be provided for in case a breakdown under these adverse conditions is to be avoided. For this reason the valve stem and other parts of the motion must be made heavy enough to sustain the stress thus imposed in case of an accident to the balancing strips and lubricating apparatus. This may be expressed by the formula:

$$R = 0.2\,Plw \tag{4}$$

in which

R = the frictional resistance of the valve,
P = boiler pressure in pounds per square inch,
l = length of valve in inches,
w = width of valve in inches.

The area through the keyway of the valve stem should, therefore, not be less than $\frac{0.2Plw}{10{,}000}$, which would put a stress of 10,000 pounds per square inch of section on the metal when in tension. In the matter of diameters the valve stem and eccentric rods must be made strong enough to carry this load in compression and for that purpose should be calculated accordingly. It must be borne in mind that this stress has to be carried back through each of the working and sustaining parts to the eccentrics, increasing somewhat as it advances on account of the added resistance of the motion of the several pieces.

Leaving the valve stem, the stress is next sustained by the rocker and its shaft, two sections of the same part that must be considered independently. The size of the arms may be calculated from the formula:

$$P = \frac{Sbh^2}{6l} \tag{5}$$

in which

P = the maximum load or resistance to be overcome,
S = the maximum fiber stress permissible in the metal used,
b = thickness of the rocker arm,
h = width of the rocker arm at any desired point,
l = length of the rocker arm from the valve stem or link block ccnnection to any desired point (see Fig. 10).

It will be noted that this formula (5) is that used for calculating the stresses imposed on a beam that is fixed at one end and loaded at the other, the expression $\frac{bh^2}{6}$ being that of the moment of resistance.

This involves the final assumption of one of the two dimensions b or h.

If, for example, b is taken to be 1½ inches and the resistance is put at 10,000 pounds, and the size of the arm is desired at 8 inches from the outer center, with a fiber stress of 10,000 pounds per square inch the formula (5) becomes

$$h = \sqrt{\frac{6 \times 10{,}000 \times 8}{10{,}000 \times 1.5}} = 5\tfrac{5}{8} \text{ inches.}$$

As the rocker shaft is usually supported by the rocker box for its entire length, it is subjected to torsional stresses only, and these are covered by the general formula:

$$P = \frac{S \pi D^3}{16 R} \qquad (6)$$

in which

P = maximum load on, or resistance to the motion of, the rocker arm,

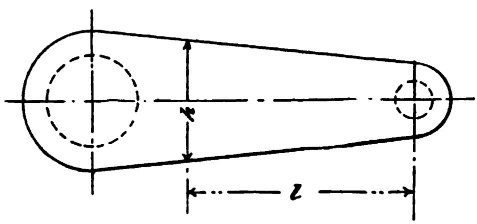

Fig. 10. Rocker Arm.

S = allowable fiber stress in the metal,
D = diameter of rocker shaft,
R = length of rocker arm,

the torsional moment of resistance being expressed by $\frac{\pi D^3}{16}$ for a round solid section.

As it is the diameter of the shaft that is to be found, the formula should be transformed into

$$D = \sqrt[3]{\frac{16 P R}{\pi S}}$$

and then substituting $P = 10{,}000$; $S = 10{,}000$, and $R = 10$ inches, we have

$$D = \sqrt[3]{\frac{1{,}600{,}000}{31{,}416}} = \sqrt[3]{51} = 3\tfrac{3}{4} \text{ inches, approximately.}$$

The link should be strong enough to sustain the thrust of the link block when unsupported by the saddle. The formula used is a modification of the general formula for a beam fixed at the ends and loaded in the middle and is

$$P = \frac{4 S b h^2}{6 L} \qquad (7)$$

in which

$P =$ the stress imposed by the valve,

$b =$ width of link across the face,

$h =$ thickness of metal in link,

$L =$ length of slot.

If we take $S = 10{,}000$; let $L = 15$ inches; and fix the width at $3\frac{1}{2}$ inches, the formula for the thickness h becomes

$$h = \sqrt{\frac{6PL}{4Sb}} = \sqrt{\frac{900{,}000}{140{,}000}} = 2\frac{1}{2} \text{ inches.}$$

The eccentric rods have already been referred to and they can be calculated as indicated with the understanding, that owing to their position and the liability to cramping and the imposition of excessive stresses due to looseness of the parts it is well to give them a strength capable of resisting a stress 25 per cent in excess of the calculated resistance of the valve.

Finally as to the eccentrics the only point to be covered is the width of the face. The diameter is usually fixed by the diameter of the axle, the throw, and the constructional requirements of the type of eccentric that is to be used. The width of the face, therefore, is determined solely by the amount of pressure that it is decided to put upon it per square inch of area. As in the case of all other bearings, an ample surface is a good investment and will repay in immunity from hot and seizing straps in the future operation of the machine. It is well, therefore, to limit the pressure to 250 pounds per square inch measured by a multiplication of the diameter by the width of the face.

This, then, closes the outline of the work to be done in the calculation of the dimensions of the several parts of the valve motion and it remains to examine the methods of application and the modifications that will have to be made in order to adapt the formulas to the two locomotives under consideration.

CHAPTER III.

DESIGNING THE VALVE MOTION.

The principles upon which the several parts of the valve motion are designed having now been set forth the next step will be the study of the application of these principles to the two engines that we have under consideration. In this work the start is made at the engine cylinders where the point affecting the whole of the valve motion is to be found in the width of the steam ports. It has been stated that the area of these should be from 1/10 to 1/12 that of the cross section of the cylinder. By referring to the cylinder drawings it may be assumed that we find that for the consolidation engine, with a diameter of 21 inches, the port measures 18 inches by 1½ inch, giving it a ratio of one to 12.8 to the cylinder section. As the piston speeds on this

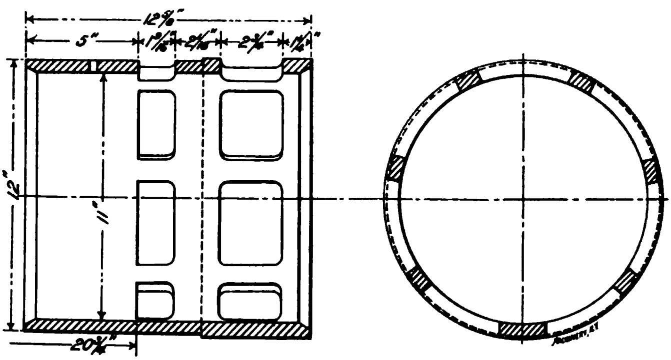

Fig. 11. Valve Chest Bushing for Atlantic Type Locomotive.

engine are to be comparatively low, and as it is always desirable for constructional reasons to keep even dimensions at all times, this variation is allowable.

In the case of the Atlantic type engine, we assume that the drawing shows that the port opening in the cylinder casting is 2¼ inches wide. This is evidently too wide and is designed for the use of a bushing to be pressed into the interior. The use of a bushing for piston valves has the three-fold advantage of making the casting of the cylinder ports easier, of facilitating the cutting of the admission ports to the exact size, and of making it possible to renew the steam chest without reboring it. Accordingly bushings like that shown in Fig. 11 are pressed into the space for the valve. In this the width allowed for the port is 1 9/16 inch, and it extends entirely around the bushing except for six bridges 1 inch wide and one 2 inches wide at the bottom. There is also a space surrounding the whole port opening in the cylinder

casting by which steam can flow around the bushing into the cylinder. The diameter of this space is about 16 inches and it is 2¼ inches wide. The outside diameter of the bushing is 12 inches. The available opening for the flow of steam with this bushing in position is, at a maximum $(1\ 9/16 \times 12) + (16 - 12) \times 2\frac{1}{4} = 27\frac{3}{4}$ square inches.

In the case of this engine a bushing is used in the cylinder as well as the valve case, and the inside diameter of this bushing is 19½ inches. This makes the ratio of port opening to cylinder area very nearly as 1 to 10.75, from which it will appear that due allowance has been made for the difference in the maximum piston speed of the two engines. The ratio of length of port to width is approximately 12 to 1. In the case of both of these engines the width of the ports is about 1½ inch, so that the travel of the valve should be 5¼ inches. Taking

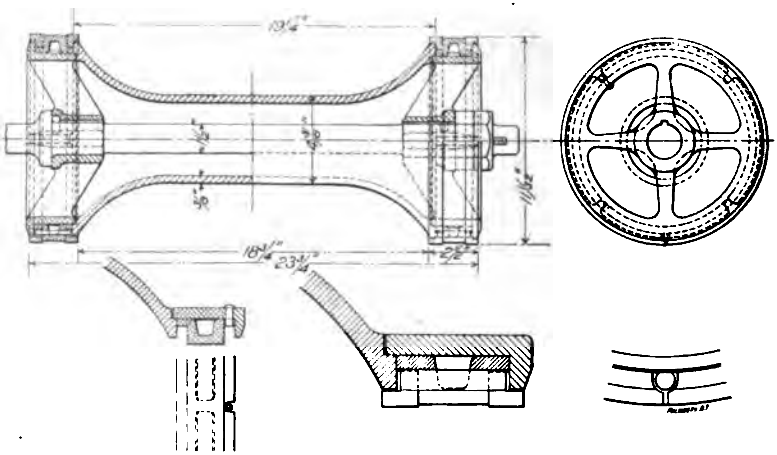

Fig. 12. Piston Valve for Express Passenger Locomotive, Atlantic Type.

this figure for the Atlantic type engine, the valve diagram, Fig. 13, can be constructed according to the directions already given.

Before starting on this it may be stated that, in the case of high speed engines, it is frequently more troublesome to get the exhaust steam out of the cylinder than it is to get the live steam into it. It is, therefore, common and approved practice to give the valve a negative exhaust lap. That is to say, the valve, when in its central position, leaves both ports uncovered to a small extent on the exhaust side. This makes the exhaust opening a little earlier than would otherwise occur and so hastens the outflow of steam and cuts down the back pressure. For this reason, the valve on this Atlantic type engine is given 1/16 inch negative exhaust lap.

As certain points in the designing of the valve motion must be decided arbitrarily, we will assume a negative exhaust lap of 1/16 inch, a lead in full gear of 1/16 inch and a maximum point of cut-off of 0.83 of the stroke. In Fig. 13 draw the circle *ABC* with a diameter of 5¼

inches, the assumed travel of the valve, and at *A* describe the lead circle 1/16 inch in radius. Draw the diameter *AC* and lay off *D*, making *AD* 0.83 of *AC*. From *D* erect the perpendicular *DE* and from *E* draw a line tangent to the lead circle. From the center *O* draw a circle tangent to the line *AE*, and its radius will be equal to the lap of the valve which will be found to be 1 inch in the present instance. By drawing *OF* at right angles to *AE* and the arc of the port opening *IH*, the angle of full port opening is obtained. In like manner the other elements of the motion of the valve may be studied. We have determined from this that the steam lap of the valve should be 1 inch.

In designing the valve, which is intended for inside admission, we take the distance between the ports on the bushings (Fig. 11) which is 20¾ inches, allowing 1 inch for lap on each side, making 18¾ inches as the distance between the lips of the valve. The distance between the outside edges of the ports is 23⅞ inches, and allowing 1/16

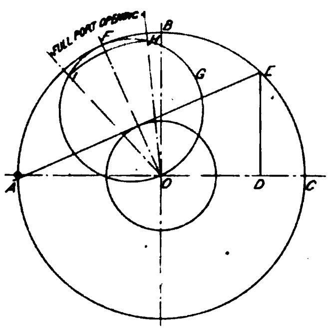

Fig. 13. Valve Diagram for Atlantic Type Locomotive.

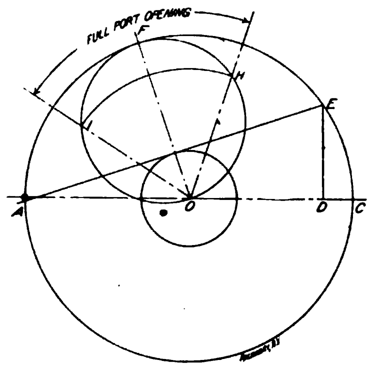

Fig. 14. Valve Diagram for Consolidation Locomotive.

inch for negative lap at each end makes the over-all length of the valve 23¾ inches.

With these dimensions it is possible to construct the valve shown in Fig. 12. In this the body is of cast iron with spring packing rings, the former being hollow so as to secure a perfect balance and permit the exhaust steam to escape through the center and flow out of the passages at each end of the steam chest. The packing can be of any desired type, that here shown being sprung in and turned 1/32 inch larger than the bore of the steam chest.

In the case of the flat valve of the consolidation freight locomotive, similar assumptions must be made, but they must be based upon different conditions of service. In the first place the engine is to be worked more slowly, and it must be able to exert its maximum tractive effort. For these reasons it is desirable that the maximum point of cut-off should be later and the period of full port opening longer. Hence it will be found to be advisable to increase the travel of the valve which may well be brought up to 6 inches and by putting the maximum point of cut-off at 9/10 the stroke, the lap of the valve, with

1/16 inch lead in full gear, becomes ⅞ inch. The diagram, Fig. 14, which corresponds to Fig. 13 for the express engine, shows the features desired. It is given in order to illustrate the difference that will be found in the diagrams of high- and low-speed engines. From this the lap will be found to be ⅞ inch, and this with the dimensions we assume as taken from the cylinder drawing makes the outside dimensions of the flat valve 21 inches. On engines of this character negative exhaust or inside lap is unnecessary, and the valve is made line and line. It may be balanced in any desired manner in accordance with the general proportions already laid down. A valve proportioned to meet these requirements is shown in Fig. 15.

It may be noted here that the slide valves should be made of hard cast iron and of a size suited to meet the conditions of the steam ports. It will be found that on ordinary engine service the outside

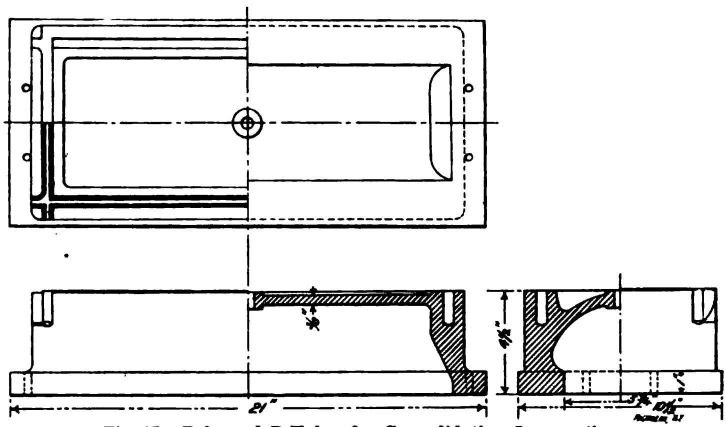

Fig. 15. Balanced D-Valve for Consolidation Locomotive.

or steam lap will vary from ⅞ inch to 1¼ inch, dependent upon the service required and the capacity of the cylinder. The lead allowed does not ordinarily exceed 1/16 inch in full gear, and is often made line and line especially when the construction is such that the eccentric rods are less than 4 feet long. The negative exhaust lap varies from line and line to ⅛ inch for high-speed engines.

The balancing of the slide valves is of special importance and the parts should be well and accurately made. Two general methods are in use, of which the oldest is known as the Richardson. It consists of ½ inch by 1½-inch strips set in suitable grooves and resting on springs, thus forming a rectangular enclosure on the top of the valve and bearing against a smooth balance plate above the valve.

A later form is the American balance which consists of a conical ring cut through at one point, and fitted to a taper bearing on top of the valve. A cover piece similar in section to the Dunbar L-section ring is employed to cover the joint. This packing requires no springs since its reaction on the taper bearing due to its elasticity and the

steam pressure tend to lift it; hence the upper part of the ring bears against the slides upon the balance plate in the same way as the Richardson valve.

The piston valve is usually made with a hollow cast iron body with followers and packing rings of an L-section at each end, and is worked inside a bushing through which the steam ports are cut as previously described. The L-shaped rings possess an advantage over square rings in that they give a positive opening and closing edge, whereas this is not necessarily the case with the latter, because the projection of the follower which holds them in position interferes somewhat with the free passage of the steam, if not taken into account as a part of the lap; while it is liable to leak, causing wire-drawing and excessive preadmission if it is so considered. On account of the many bridges on the ports of the bushing and the more or less checked flow of steam past the top of the valve towards the cylinder, the diameter ot

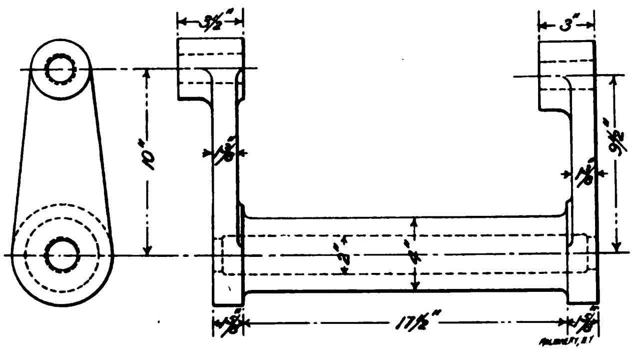

Fig. 16. Direct Motion Rocker Arm for Atlantic Type Locomotive.

the piston valve should be such that circumferential port should be at least 50 per cent longer than those for a flat slide valve. Further, the valve should invariably be made for inside admission, so that the stuffing-boxes are not obliged to sustain boiler pressure, and the valve events can be made the same as with the ordinary slide valve.

With the proportions of the valve thus worked out, the designing of the other ports of the motion becomes a comparatively simple matter. Before this can be done, however, it is usually necessary to locate some of the other parts, work out the frames and driving mechanism so as to insure proper clearances when the engine is in motion. Such work usually decides the location of the link, rocker-boxes and connections. As would be clearly shown in a side elevation of the engines, the arrangement of the working parts of the two are quite different. In the case of the Atlantic type engine, the valve has a direct motion from the eccentrics obtained by the use of a rocker like that shown in Fig. 16. From this it will be seen that the valve arm is longer than the eccentric arm, so that the throw of the latter is less than the travel of the valve. According to the proportions given, the throw of

the eccentric should be 4.98 inches. This can be made 5 inches, which will be done.

In the case of the flat valve of the consolidation engine, an indirect motion rocker like that shown in Fig. 17 is used. Here, too, it has been found to be convenient to make the eccentric arm shorter than the one driving the valve stem. With the proportions chosen, the throw of the eccentric becomes 5.08 inches, and it is made 5 inches as before, thus modifying to a very small extent the several valve events as found from the diagram, but not enough to materially affect the action of the engine.

With the valve and the eccentric proportioned, it becomes possible to work out the details of the whole motion. Starting with the eccentric, its diameter is dependent, to an extent, upon the diameter of the

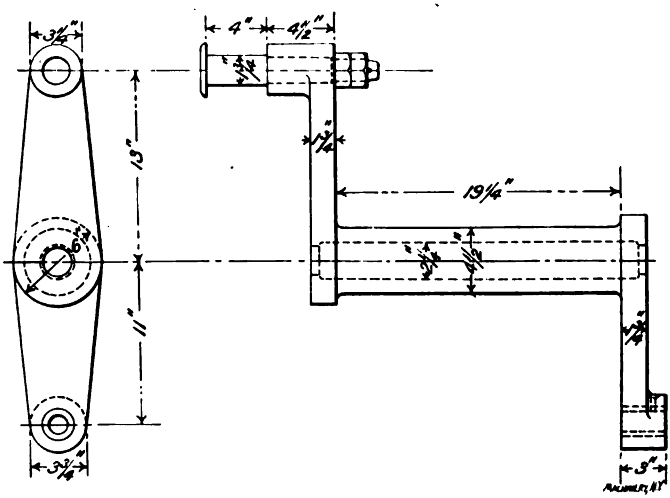

Fig 17. Indirect Rocker Arm for Consolidation Locomotive.

axle. The form used depends upon the taste of the designer. On many roads solid eccentrics are in use that must be put in place before the wheels are pressed upon the axle. In some cases the two eccentrics are made in one piece, but the general practice is to make the eccentric in halves and of cast iron, securely bolted together and keyed to the axle. For this detail there are a number of designs, one of which is shown in Fig. 18. It is made as small as possible, consistent with strength. If the axle is assumed to be 9 inches in diameter, at least 1 inch of metal should be allowed on the thin side, which would make the outside diameter 16 inches; the allowance for re-turning makes 16¼ inches. The width of the eccentric must be sufficient to permit of keying without tilting, and bolting the two parts together without danger of splitting or cracking.

We have seen that the eccentric must have ample strength to drive the valve under the most adverse conditions when the packing strips

are broken and the surface dry; it must also have ample bearing area to prevent heating. Substituting the values of the valve in formula (4) we have a boiler pressure (P) of 200 pounds; a length (l) of 21 inches, and a width (w) of 10½ inches. Hence the resistance

$$R = 0.2 \times 200 \times 21 \times 10.5 = 8{,}820 \text{ pounds.}$$

As the pressure on the bearing surface should not exceed 250 pounds to the square inch, the area of this surface should not be less than

$$\frac{8{,}820}{250} = 35.3 \text{ square inches.}$$

As the diameter is 16¼ inches, the width should not be less than 2 3/16 inches. As an increase over this is desirable, and as there is plenty of room, it is made 2¾ inches, which cuts the pressure down to

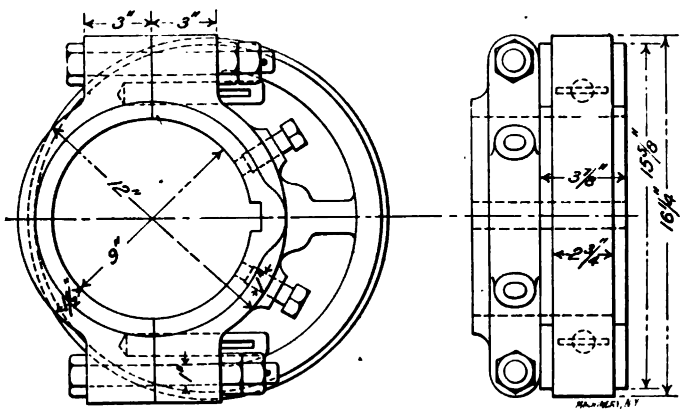

Fig. 18. Eccentric for Consolidation and Atlantic Type Locomotives.

about 200 pounds, thus giving good working conditions so far as bearing surface is concerned even under exceptional resistances.

The eccentric is keyed to the axle, as the adjustment of this part and fastening by setscrews are things of the past and are entirely obsolete methods of construction. The details of the form shown here are sufficiently distinct in the engraving to require no explanation. Attention is merely called to the fact that every precaution in the way of check nuts and cotters is used to prevent the parts from becoming loose or lost.

Closely allied with the eccentrics are the eccentric straps. They may be of cast iron or bronze and must have a strength sufficient to move the valve under adverse conditions without an appreciable amount of yield. This is necessary in order that they may preserve their full bearing surface in contact with the eccentric at all times and not pinch the latter because of some distortion or yielding. The formula (7), which is the general one for a beam fixed at the ends and loaded in the middle, may be used for the calculation of the body of the strap. It may be modified, however, to

$$P = \frac{4\,Q\,S}{l} \qquad (8)$$

in which

P = the stress imposed by the valve,

S = allowable fibre stress in the metal,

l = distance between fastening bolts in inches,

Q = the section modulus.

The latter must be worked out for all sections other than a rectangle which is $\frac{b\,h^2}{6}$ as already given. In the case of the strap shown in Fig. 19, the width is 3¾ inches and the depth 2⅞ inches. The section is flat on one side and semi-circular on the other, and if the computa-

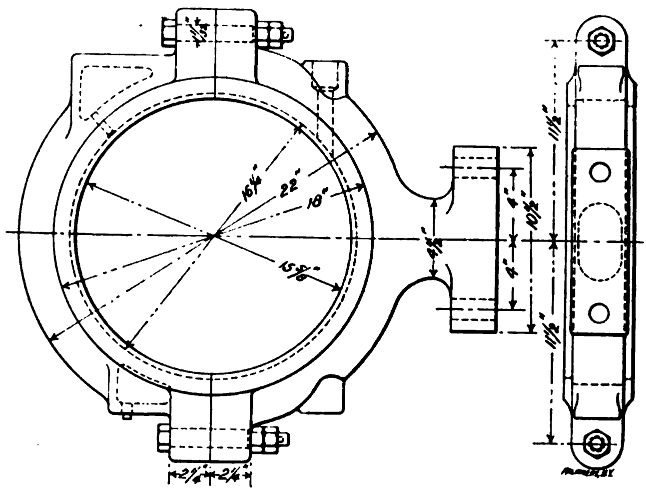

Fig. 19. Eccentric Strap for Consolidation and Atlantic Types of Locomotives.

tions are made as to its strength it will be found that the maximum fiber stress put on the metal will fall below 2,500 pounds per square inch of section. This is low, but it is one of those places where plenty of metal is a good investment as it insures against hot straps and annoying delays upon the road when the engine is in service. The neck of the strap is also made of ample strength and the foot is faced to receive the direct thrust of the rods. The bolts used for the fastenings should be of ample size to hold the parts firmly together, and when this is done their strength will be sufficient to carry the load that is put upon them. The fastenings of the eccentric rods to the straps is made much more secure than it was when engines and valves were lighter. At that time there was a possibility of adjusting the length of the rods, with the result that they frequently slipped in service. Current practice does away with this adjustability with a resultant simplification of the parts.

Fig. 20 shows a substantial form of eccentric rod that is used on

the engines under consideration. It will be seen that it is made in "rights" and "lefts" so that the jaws at the forward end line up together to take the link. It is of the utmost importance that these rods should be exceedingly stiff and rigid in order to prevent springing when they are working under compression. If that occurs the action of the valve is not what it should be, and an extra and unnecessary stress is put upon the bolts, pins, links and eccentric straps, all of which tends to a more rapid wear and an increase of the cost of maintenance, to say nothing of the danger of causing delays and breakdowns on the road.

The distance from the center of the axle to the rocker arm, or the radius of the link, is 49 inches in the case of the consolidation engine

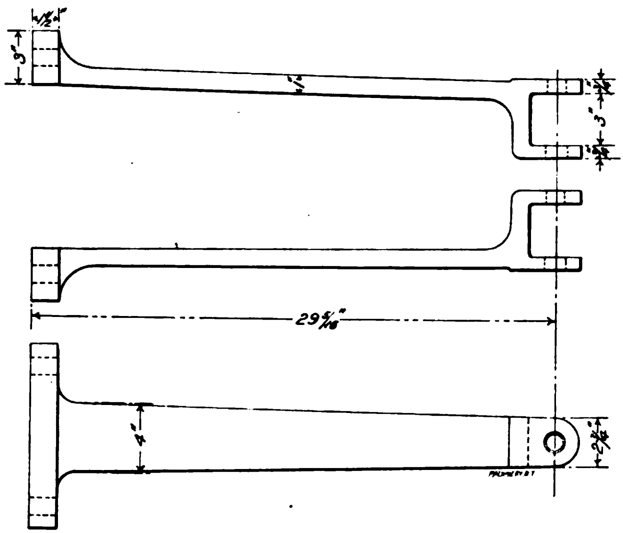

Fig. 20. Eccentric Rod for Consolidation Type of Locomotive.

and 60 inches for the Atlantic type. This causes a variation in the length of the eccentric rods of from 29 5/16 inches to 40 5/16 inches. As these rods are flat they must be calculated accordingly. We have seen that the probable maximum load imposed by the valve will be 8,820 pounds. In order that there may be an ample margin of strength, it will be well to take this load at 10,000 pounds and proportion the parts accordingly. By using the following formula for this purpose, and taking the length of the road at 30 inches and the thickness as 1 inch, we have

$$\frac{P}{A} = \frac{S}{1 + \frac{q\,l^2}{r^2}}$$

in which

P = resistance of the valve = 10,000 pounds,

A = area of rod = width of rod $\times$ 1,
S = allowable fiber stress = 10,000 pounds per square inch of section,
l = length of rod = 30 inches,
$r^2 = 1/12$,
$q = 0.00016$.

By substitution and transposition this formula becomes

$$A = \frac{(1 + 0.00016 \times 900 \times 12) \times 10{,}000}{10{,}000} = 2.73.$$

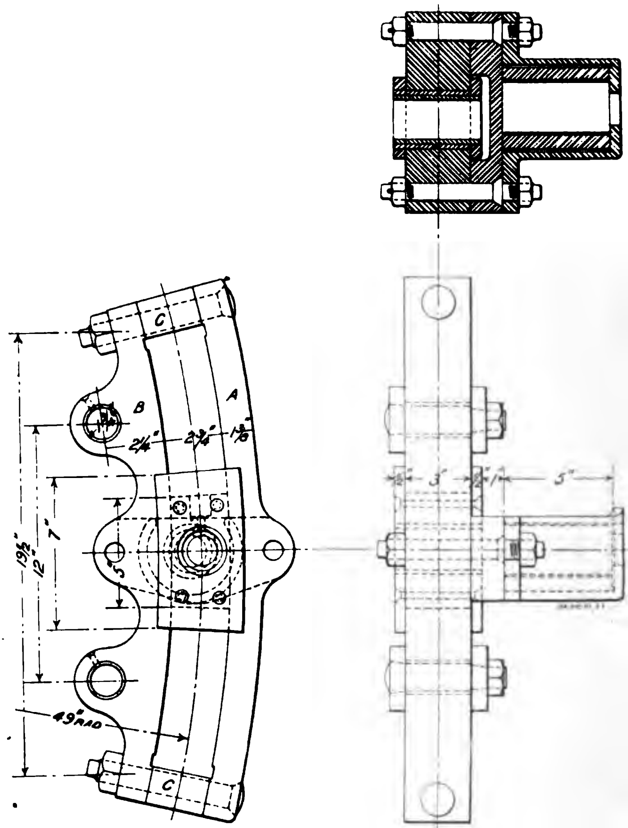

Fig. 21. Wrought Iron Case-hardened Link for Consolidation Locomotives.

The rod can, therefore, be made 2¾ inches wide at the small end and tapered to any desired width (4 inches in this case) at the large.

The diameter of the valve stem can be calculated by the same formula by changing the value of r^2 to $\frac{d^2}{16}$, making $A = \frac{\pi d^2}{4}$, and making l = about 85.

By substitution and transposition, the formula becomes

$$\frac{4\,P}{\pi\, d^2} = \frac{10{,}000}{1 + \dfrac{(0.00016 \times 7225)16}{d^2}}, \text{ and finally}$$

$$d = \sqrt{\frac{2 + \sqrt{74\,\pi + 4}}{\pi}} = 2.35,$$

so that the valve rod may be made about 2⅜ inches in diameter.

The link is also a part that needs most careful attention not only in

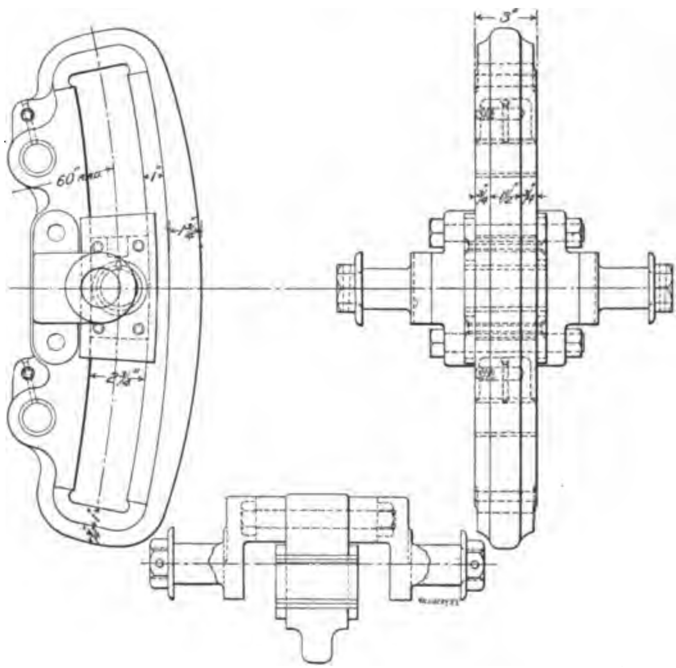

Fig. 22. Cast Steel Link for Atlantic Type Locomotive.

the designing and proportioning, but in its manufacture. For many years the links on American engines were made of wrought iron, casehardened and carefully ground to truth. Of late cast steel has been introduced with good results as far as operation is concerned and with a considerable saving in first cost. This metal also possesses the advantage of permitting a better distribution to carry the loads, and can, therefore, be made lighter.

The common form of skeleton link is shown in Fig. 21. It is adapted for use on the consolidation locomotive and owing to limits of space it has been made a trifle shorter than the general proportions given. With an eccentric throw of 5 inches the distance between the

eccentric rod pins should be 12½ inches, but in this case it is made 12 inches. The link is built up of four parts, the front *A*, the back *B*, and the two fillers at the ends *CC*. The holes for the eccentric rod pins are protected by casehardened bushings, and the block is given an ample bearing surface 5 inches long. As in the case of the eccentric rods the link must be of ample strength to do the work without springing. For this we may use formula (7) though it is common practice to depend upon the supporting of the saddle to prevent springing. Then, again, as the load of the unlubricated, unbalanced valve is excessive, and as it is desirable to make the link as light as possible so that it may be easily handled in reversing and on account of the support of the link saddle the stress allowed for may be dropped to 5,000 pounds instead of making it 10,000, and the allowable fiber stress may be raised to 12,000 pounds. This leaves the link with ample strength for the ordinary working while in the case of an accident it will not be overstrained on account of the length of the bearing of the link block. The formula as thus modified becomes:

$$P = \frac{4\,S\,b\,h^2}{6\,L}, \text{ in which}$$

P = load imposed by the valve = 5,000 pounds,
b = width of the link across the face = 3 inches,
h = thickness of metal in the link,
L = half the length of the slot = 9.75 inches,
S = allowable fiber stress = 12,000 pounds per square inch section,

$$h = \sqrt{\frac{5000 \times 6 \times 9.75}{4 \times 12{,}000 \times 3}} = \sqrt{2.03} = 1.42$$

from which the thickness may be made 1 7/16 inch.

In case of the cast steel link shown in Fig. 22 for the passenger locomotive, a similar course of reasoning can be followed, except that the fiber stress put upon the metal should be kept down to 10,000 pounds, or even less. It will also be well to work out the section modulus as this has an important bearing on the rigidity of the link.

The method of calculating the size of the rocker has been provided for in formulas (5) and (6).

Taking the rocker for the consolidation engine as shown in Fig. 17, if the valve resistance is placed at 10,000 pounds, the length of the arm at 13 inches and the thickness is assumed to be 1¾ inch, then using formula (5)

$$P = \frac{S\,b\,h^2}{6\,l}, \text{ in which}$$

P = valve resistance = 10,000 pounds,
S = allowable stress of metal = 12,000 pounds,
b = thickness of arm = 1.75 inch,
h = width of arm,
l = length of arm = 13 inches.

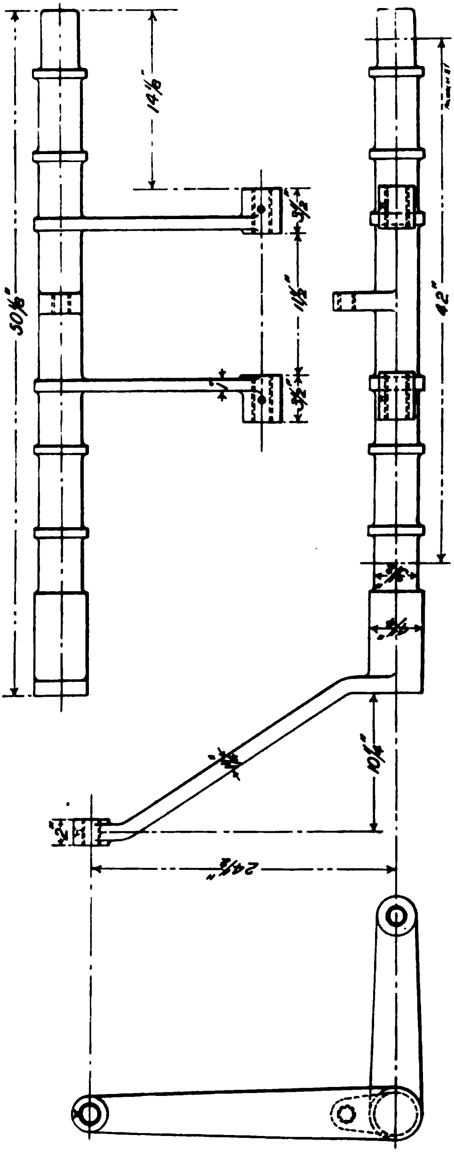

Fig. 23. Lifting Shaft for Consolidation Locomotive.

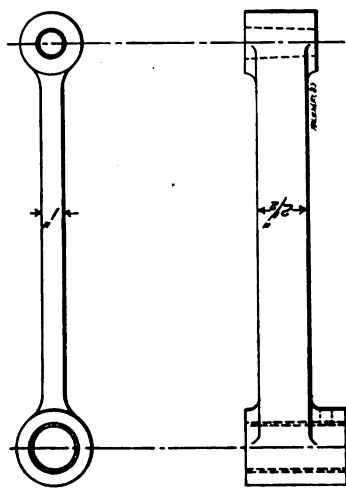

Fig. 24. Link Hanger for Consolidation Locomotive.

By transposition and substitution this becomes

$$h = \sqrt{\frac{6 \times 18 \times 10{,}000}{1.75 \times 12{,}000}} = \sqrt{37} = 6 \text{ inches.}$$

As it is always well to have an ample bearing surface for the rocker, and as lightness is a desirable quality in the moving parts, cast steel is used as a metal and the bearing is made hollow. By assuming a core of 2¼ inches and deducting the value of the metal thus removed from the strength of the shaft, this core would be able to carry a load

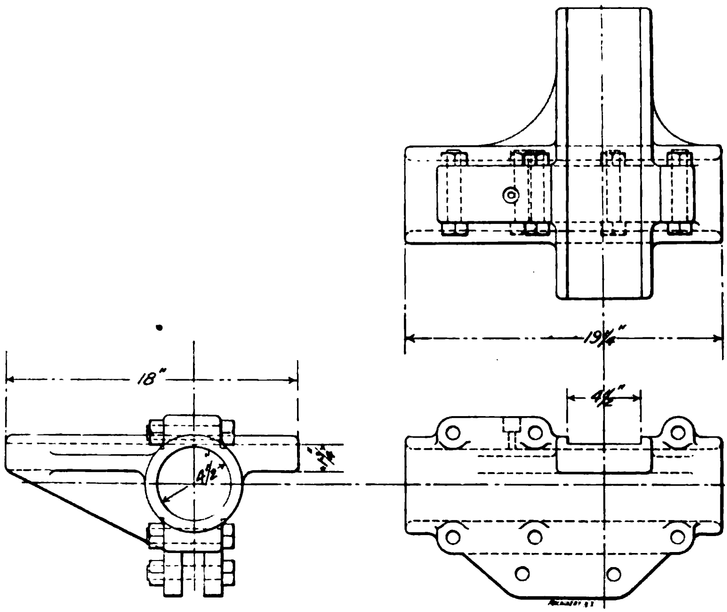

Fig. 25. Rocker-box for Consolidation Locomotive.

of about 1,750 pounds if calculated to formula (6). This load is then added to the actual valve pressure, making $P = 11{,}750$ pounds.

$$D = \sqrt[3]{\frac{16\,P\,R}{\pi\,S}}, \text{ in which}$$

$D =$ diameter of rocker shaft,
$P =$ valve resistance $= 10{,}000 + 1{,}750$ pounds,
$R =$ length of rocker arm $= 13$ inches,
$S =$ allowable fiber stress $= 10{,}000$ pounds.

By substitution the formula then becomes

$$D = \sqrt[3]{\frac{16 \times 11{,}750 \times 13}{31{,}416}} = \sqrt[3]{77.8} = 4.27 \text{ inches.}$$

In this case the diameter is made 4.5 inches. These are the outlines of

the methods to be followed in the proportioning of the parts of the valve motion.

The lifting-shaft (Fig. 23) and the link-hanger (Fig. 24) are important parts that must be carefully designed and located. In the matter of strength, if they are made stiff enough to do the work that is all that is required, and their only load is to carry the weight of the link and resist the forces due to the angularity of the link and the binding action of the block in its slip. They are, however, closely associated with the proper location of the saddle-pin on the link, for on the proper combination of dimensions and positions of the saddle-pin, hanger, lifting-shaft and box depends the smooth and correct action of the valve. All parts of the valve motion may be carefully and accurately laid out and if the bearing of the lifting-shaft is improperly located, the action of the valve will be defective.

An outline of the method to pursue is to take a template of the link (full size preferred) and locate it in the extreme positions of mid-gear. On transverse lines through the centers of the two positions of the templates locate points equally distant from the center which coincide with the arc swept through by the lower end of the hanger when the lifting shaft is in mid-position. These points will indicate the proper position of the saddle pin. The lifting-shaft box should be so adjusted that the lower end of the hanger will sweep through the corresponding position of the saddle-pin when the link is raised and lowered to full backing and forward gears respectively. The designer should make himself familiar with all of the vagaries and peculiarities of the Stephenson valve motion, and this involves a careful study both of what has been published and of the working out of the problems on the drawing board.

It only remains now to call attention to the form of rocker-box that is used; that intended for the consolidation locomotive being shown in Fig. 25. These boxes are usually bolted to the guide yoke, are made of cast iron and afford a support for the rocker bearing throughout its whole length. All pin holes of the working parts of rockers, link and rods are protected by casehardened bushings.

With this outline of the course to be followed in working out the Stephenson link motion the reader is cautioned against trusting to any haphazard methods of design and recommended to master its intricacies in every particular before attempting to make a practical application of a design to a locomotive.

CHAPTER IV.

THE WALSCHAERTS VALVE MOTION.

Up to the present time, whenever an American has considered the designing of a locomotive, the work has invariably been associated with the use of the Stephenson link motion for the operation of the valves. This is true with the exception of a very few instances where interested parties have had some special design of gear to exploit. That the Stephenson gear has held its own for so many years speaks well for its efficiency, and indeed it has been found that, in the matter of steam consumption, in special cases it holds this figure down to within a very small percentage of the best that can be obtained with the Corliss gear.

On the continent of Europe, on the other hand, the Walschaerts gear or a modification thereof is almost exclusively used, and it is claimed to possess many advantages over the Stephenson motion, the most prominent of which is the maintenance of a constant lead for all points of cut-off, an advantage that is not universally acknowledged, however. There are indications now, however, that the Walschaerts gear will receive a more and more extensive application in this country, hence it is well to discuss and analyze it in considering the designing of a locomotive.

The reason for this change of attitude regarding the Walschaerts gear is due to a number of difficulties that have been experienced with the Stephenson valve motion on the large and heavy locomotives of modern construction. Among these are the excessive wear of the heavy eccentrics and straps, and the large amount of space between the frames that is occupied by the eccentrics, rods, links, and hangers, making it exceedingly troublesome or quite impossible to properly brace the frames; the Walschaerts gear has, therefore, been very successfully applied to a number of engines of recent design.

It is more accessible than the Stephenson motion in that it is applied outside the wheels and requires only a single eccentric return crank and a connection to the crosshead for its operation. The eccentric crank may be a comparatively small pin attached to the main crank, and it does the work of the two heavy eccentrics of the ordinary gear. This arrangement leaves the entire space between the frames clear for the bracing of the same.

Further, it produces a more uniform steam distribution with a lower percentage of pre-admission, to which is added a constant and moderate amount of lead for early cut-off, though on the resultant economy in steam consumption there can be but slight difference when the gears are both in first-class condition. Finally, it is not so likely to get out of order as when the driving is done by eccentrics; since, when the parts

have been properly fitted, they are not liable to get out of place except when damaged by collision or other accident to which they are more exposed.

The Walschaerts gear is, in reality, much simpler to understand than the Stephenson. Of course, since the results of the two are nearly identical, each must have parts whose functions correspond to those of the other. In the case of the Walschaerts gear the valve receives its motion from two sources, the crosshead and an eccentric crank whose center is located 90 degrees from the center line on the main crank, when the center lines of the cylinder and gear motion coincide and pass through the center of the axle.

By referring to Fig. 4 and the description accompanying it, it will be seen that, as the crank stands on the center, the eccentrics at *C* and *D* are given an angular advance which is equal to the sum of the lap and lead of the valve in full gear. In the case of the Walschaerts

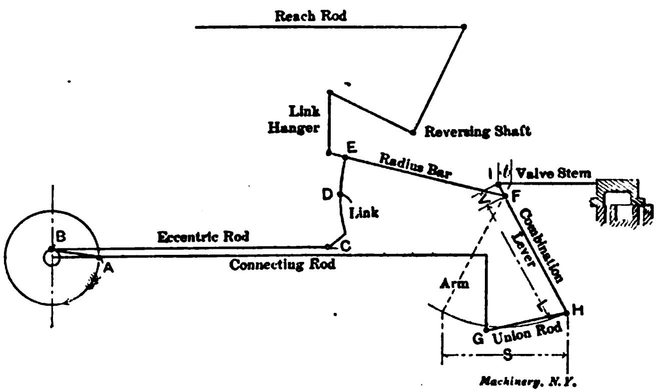

Fig. 26. Diagram of the Walschaerts Valve Gear.

gear, which is shown diagrammatically in Fig. 26, when the crank is on the forward center at *A*, the center of the eccentric is at *B* at right angles to the crank, provided the point *C* is close to the center line of the cylinder. From *B* an eccentric rod runs to *C*, the lower end of a fixed line *CDE* which is pivoted at *D*. This link has a groove in which the link block attached to the radius rod slides up and down. The length of this rod from *E* to *F* is equal to the radius of the link itself. If now the valve stem were to be connected directly to the end of the radius bar, it is evident that when the crank is on the center, the valve will be in its central position either for forward or backward motion and for any position of the link block of the radius bar in the link. Under these circumstances, there could be no lap or lead to the valve.

The lap and lead is arranged for by dropping a rigid arm down on the crosshead to *G* and from this a union rod is led out to the lower end of the combination lever at *H*, this lever being pivoted at the end of the radius bar at *F* and extending up to the valve stem connecting

at *I*. It is evident that the inclination of the combination lever will be the same at the end of the stroke regardless of the position of the radius bar; and that, therefore, the horizontal displacement of the point *I* and the valve stem will be the same on either side of a vertical line through *F*. This horizontal displacement is equal to twice the sum of the lap and the lead, hence the latter is constant for all points of cut-off. These same statements hold true for the opposite end of the stroke, when *A* is on the back center and *B* at the bottom, barring a slight variation due to the angularity of the rods.

In the case shown, where the eccentric *B* follows the crank the engine runs ahead when the radius block is at the top of the link and backwards when it is at the bottom. These conditions would be reversed by having the eccentric lead the crank. The crosshead, then, imparts a motion to the valve equal to the lap and lead when the crank is on either center, just as the angular advance of the eccentrics does in the case of the Stephenson gear.

It will be noticed that the eccentric and the crosshead tend to move the valve in opposite directions during the first half of each stroke and in the same direction during the last half; or, in other words, they work in opposite directions during the first and third quarters of a revolution of the crank starting from either dead point, and together during the second and fourth quarters. The motion derived from the crosshead is constant and is not subjected to reversal in the reversing of the motion of the engine, which is done exclusively by a change in the motion imparted by the eccentric, which also controls the variation of the points of cut-off.

In order to accomplish this the motion of the eccentric is transmitted through an oscillating link pivoted at its center and so slotted that a link block attached to the back end of the radius bar can be moved through its whole length, and by placing this above or below the center, a reversal of the engine will be obtained. This motion, either direct or indirect, is taken up by the radius bar and carried out to the combination lever, where it is combined with that obtained from the crosshead and the resultant imparted to the valve. The motion is therefore the same as though it were derived from an eccentric the center of which could be moved on the line of a chord across the axle from one extremity to the other of the throw.

It is evident from this that the several connecting points along the combination lever bear a most important relationship to each other, which must be maintained in order that a proper movement of the valve may be obtained.

The distances between these several points may be found by the formula

$$S : t = L : V, \text{ or } V = \frac{L\,t}{S} \qquad (9)$$

in which

$S =$ stroke of piston,

$t =$ twice the sum of the lap and lead,

L = distance between the crosshead connection H and that of the radius bar F, Fig. 26,

V = distance between connection of the radius bar F and that of the valve stem I, Fig. 26.

For an outside admission of steam, as in the case of the ordinary slide valve, the connection F, Fig. 26, falls below that of the valve stem so that the crosshead increment of the motion is the same as though it were derived from an eccentric or crank opposite the main crank; while the increment controlling the direction of engine rotation is derived from what amounts to an eccentric leading the crank for forward motion, just as in the case of direct-acting eccentrics in the case of the Stephenson gear.

A modification of the diagram of Fig. 26 is shown in Fig. 27, where it is arranged so that the block shall be at the bottom of the link for forward motion and at the top for backing, which is the reverse of that shown in Fig. 26, a condition that is brought about by simply

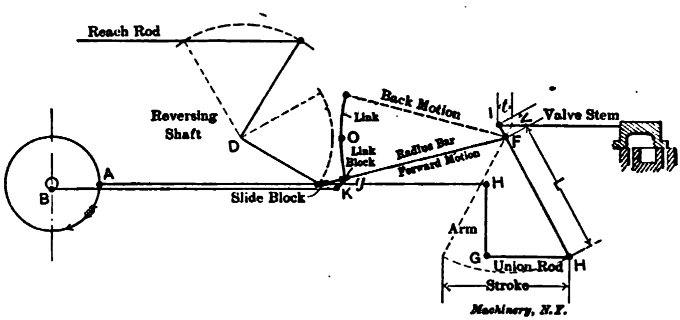

Fig. 27. Modification of the Walschaerts Valve Gear.

locating the eccentric center 90 degrees ahead of the crank instead of behind it.

When used in connection with a piston valve having an inside admission, the valve stem is usually attached below the connection F of the radius bar on the combination lever, and the eccentric follows the main crank when going ahead with the block at the lower end of the link, which results in the crosshead motion having the same effect as an eccentric on the same side of the axle as the crank.

The reversal of the motion is accompanied by means of a reversing shaft with lever and reach-rod attachments as in common use, and acting upon the radius bar either by a sliding block pivoted directly to the reverse arm, as shown in Fig. 27, or by a hanger from which the bar is suspended. In the first case the bar slides to and fro in the block, and the slip of the link block is equal to the versed sine of one-half of the arc through which the link moves. When the hanger is used it must be of sufficient length to avoid an excessive slip of the block. It is for this reason that the reversing shaft is frequently placed below the radius bar so that the swing of the hanger will tend to compensate for the oscillation of the link and thus reduce the slip

to a minimum. In this, too, it is important to ascertain the correct location of the point of suspension of the hanger and to so place the reversing shaft that the hanger does not cause irregularities in the motion of the valve by swinging the link block out of the correct alignment with the radius bar when the former is in any of the several positions that it may occupy.

Having thus reviewed the principles in accordance with which the Walschaerts gear operates, it remains to analyze the several parts and determine the relative proportions existing between them. The lap and lead motion having been obtained by means of formula (9) the radius of the eccentric crank, or the travel of the point F of the link block for any given travel of the valve is found by the formula:

$$b = \frac{R\sqrt{a^2 - c^2}}{R + c} \qquad (10)$$

for outside admission and,

$$b = \frac{R\sqrt{a^2 - c^2}}{R - c} \qquad (11)$$

for inside admission where

a = half travel of the valve,
b = half travel of the link block at point F,
c = lap and lead of the valve,
R = radius of the main crank.

From these formulas one-half of the travel of the point F can be obtained, which for the sake of simplicity and with no appreciable error can be considered as equal to one-half the travel of the link block. In this both are considered to be moving in straight lines, which is not strictly the case, but the error is insignificant and has an effect only at the extremities of the travel, which decreases as the point of cut-off is made earlier, so that for practical reasons the difference may be ignored. In fact the method of supporting the radius bar has an influence upon the horizontal movement of the link block, so that the difference between it and the point F will have to be calculated independently for each individual case and point if a mathematical precision is to be obtained, which has been found to be entirely unnecessary in practice, when the suspension points have been properly laid out.

It will be seen that, in these last two formulas, the proportions of the combination lever are ignored, because the lap and lead as obtained from formula (9) puts this in the same ratio to the main crank as the two arms of the combination lever are to each other. With an outside admission valve, the locus of the virtual eccentric may be considered to lie in a straight line at right angles to the center line of the crank but upon the opposite side of the axle center to that of the crank itself, or from formula (9)

$$R : (R + c) = L : (L + V),$$

while with an inside admission valve the sign is changed and we have

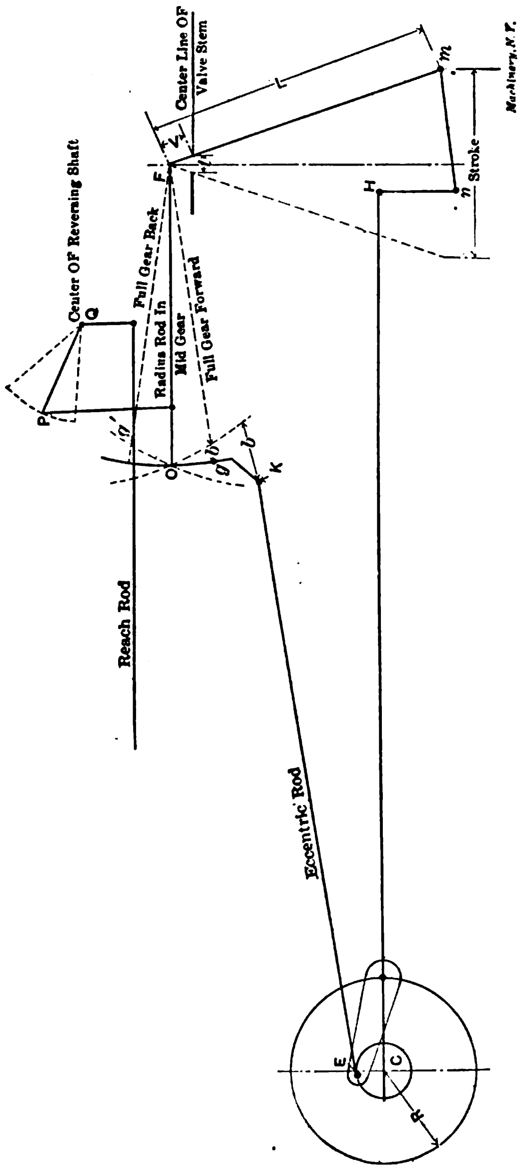

Fig. 28. Method of Reversing the Walschaerts Valve Gear.

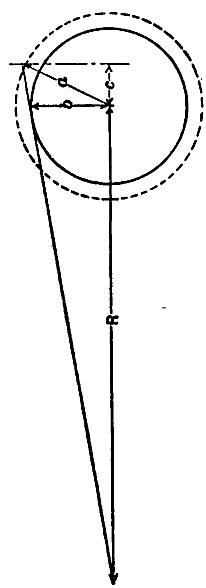

Fig. 29.

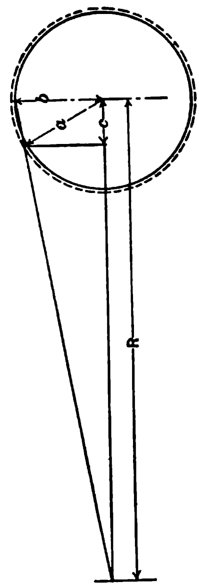

Fig. 30.

$$R : (R - c) = L : (L - V),$$

in which case the valve stem falls below the point F, which must, therefore, have a longer travel than with outside admission in order to maintain the same travel of the valve.

The location of the virtual eccentric at right angles to the center line of motion is equal to $\sqrt{a^2 - c^2}$, from which we may obtain the half travel of the point $F = b$ by the following proposition:

$$\sqrt{a^2 - c^2} : (R + c) = b : R$$

or

$$b = \frac{R\sqrt{a^2 - c^2}}{R + c}$$

for outside admission and

$$\sqrt{a^2 - c^2} : (R - c) = b : R$$

or

$$b = \frac{R\sqrt{a^2 - c^2}}{R - c}$$

for inside admission as per formulas (10) and (11).

In this case b may be considered as equal to the radius of the eccentric crank, as it would be were it possible to attach the front end of the eccentric rod to the link block. This is laid out graphically in Figs. 29 and 30. If a is the same in both cases, b will be greater in Fig. 30 than in Fig. 29.

With these formulas as a basis it is possible to proceed with the determination of the actual radius of the eccentric crank. Starting with Fig. 27 as a basis, and having settled the travel of the valve and the throw of the point F, the next thing to determine is the distance Og which the link block will have to be moved from its central position to full gear. In this there are two antagonistic requirements to be reconciled. On the side of the angularity of the radius bar, it is desirable that this movement should be as small as possible, while with the angularity of the link in view it should be as large as it can be made. Hence it is necessary to compromise between the two. Practical experience has shown that it is not well to swing the link through an angle of more than 45 degrees, so that assuming this as that to be used we have:

$$Og = \frac{b}{\tan 22\frac{1}{2} \text{ deg.}} \qquad (12)$$

in which the angle given is that of half the travel on each side of the center.

The connecting point K between the eccentric rod and the link should be as near the center line of the engine as practicable. With inside admission piston valves it frequently happens, however, that the link fulcrum O is rather high, so that a large crank radius would be required in order to secure the requisite amount of motion. Consequently good judgment must be used in order to secure practical

results. If, however, it is found necessary to locate this point K at any appreciable distance above the horizontal center line of the engine, the center of the eccentric crank should be set back with the same angularity so that a line drawn through it and the center of the axle will be at right angles to one drawn from the center of the axle to the point K.

The fore and aft position of K is a matter of importance and it should be such that it will swing through the same angle on each side of its central position at the same time compensating for the angularity of the rod.

No absolute formula can be given for the location of the point K, as it must be worked out for each individual case. It will always be back of the tangent to the link drawn through O in the central position, and the distance will depend upon the inverse ratio of the relation existing between the throw of the eccentric and the length of the eccentric rod. It is further influenced by the variation in the angularity between the center line of motion and the tangent to the link as well as its distance from the fulcrum O. Under ordinary conditions the point K will fall from 2 inches to 5 inches in the rear of a tangent to O.

With very short eccentric rods, there may be some difficulty in securing this equal augularity of swing, in which case the distance of K from the tangent to the link can be reduced somewhat and not materially affect the opening and closing points of the valve. The maximum port opening will be affected, it is true, but as this is usually more than that actually required, and as the irregularity gradually disappears as the cut-off is made earlier, it will have very little influence on the working of the engine.

Having obtained the horizontal motion of the link-block at the point g from the formulas (10) and (11) as well as the angular swing of the link, it is evident that the point K must move through the same angle; we thus have by making $k = b'$,

$$Og : OK = b : b'$$

or

$$b' = \frac{OK \times b}{Og} \qquad (13)$$

This will also be the radius of the eccentric crank, but owing to the angle made by OK with the tangent to the link, the radius will decrease as this angle increases and must be laid out in each case to meet the conditions involved.

The location of the center Q of the reversing shaft (Fig. 28) must be such that the end of the arm QP at full gear forward will be in such a position that the lower end of the hanger will swing through an arc tangent to the radius bar at its point of attachment. An exact equality in this respect is impossible to attain throughout the whole range of cut-off, so that the especial attention should be directed toward securing it over the range in which the engine is to work, which

should be between a 30 and 60 per cent cut-off in forward gear for road engines, and half and full gear, front and back, for switching engines.

It will be found, then, that owing to the unavoidable irregularities inherent in the transformation of circular into rectilinear motion (inherent in all types of valve gears) the arc through which the point P passes to the greatest extent is in one of the upper quadrants of the circle.

The crosshead connecting point n, Fig. 28, in relation to the point m at the lower end of the combination lever will also be found to have more or less effect upon the regularity of the motion, of which advantage can be taken in the laying out of the gear. Ordinarily the best position will be found by locating the point n on the same horizontal line with m when the combination lever is in a vertical position.

With the slide bearing pivoted directly on the lifting arm as indicated in Fig. 27, the movement of the end of the radius bar is practically in a straight line throughout the whole range of cut-off in both directions, a fact which greatly simplifies the construction of the gear,

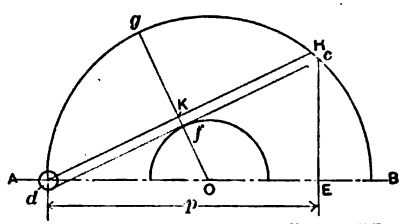

Fig. 31.

and is, for that reason, to be preferred to the use of a swinging hanger.

The valve diagram may be laid out in the same general way as with the Stephenson gear; but as there is no increase of lead for the earlier points of cut-off, a definite amount of lead must be given from the start, by which a complication is added to the formula for finding the lap for a given lead, valve travel and maximum cut-off. It can, however, be done as follows:

In Fig. 31 let AB be the travel of the valve and E the point of maximum cut-off. Draw Ec at right angles to AB and describe the semicircle AcB, on AB as a diameter. With the lead as a radius and A as a center, draw the circle d, and draw cd tangent to circle d. Draw Og at right angles to cd and draw the lap circle tangent to cd, the radius of which, Of, will be the required lap.

Further, draw AH parallel to cd and it will be the locus of the virtual eccentric center and

$$AO = a; \; OK = c$$

in formulas (10) and (11).

Zeuner's diagram is as applicable here as in the case of the Stephenson gear with the difference that the locus of the virtual eccentric

is a straight line instead of a curve, and the points R^1, R^2, R^3, R^4 of Fig. 9 determine the radii for different travel circles of the cut-off as well as the corresponding diameters of their respective valve circles.

As in the case of the designing of the Stephenson valve motion, it is necessary to make some arbitrary assumptions in order to determine the dimensions of the several parts by the use of the formulas that have been developed. This is done in connection with the diagram of Fig. 28. In this we have certain dimensions already given. The first is the stroke of the piston which is 26 inches. By taking the travel of the valve at 5¼ inches and its maximum point of cut-off at 0.83 of the stroke as in the case of the Stephenson gear and fixing the lead at ⅛ inch the valve diagram of Fig. 32 can be constructed from which the lap of 1 inch will be obtained. The sum of the lap and lead will then be 1⅛ inch. As the half stroke of the crosshead is 13 inches the ratio of the lap and lead to this motion will be as 1 to 11.55.

By first laying off an outline of the main crank, the center line of the valve stem, the connecting rod and crosshead and a vertical line

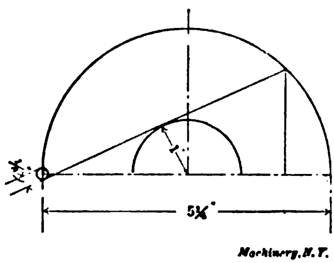

Fig. 32.

passing through the center of the connection between the valve stem and the combination lever when it is in the central position, the proportions of the latter are first obtained. The length of the long arm should not be less than 2¼ times the stroke, if too great angularity of motion is to be avoided. In this case the attachment of the valve stem will be below the point *F*. If we assume the distance between the two to be 3½ inches then the total length of the rod becomes 40.415 inches or a little more than 40 13/32 inches. Laying off the lap and lead on the center line at *t*, the position of the combination lever at a forward end of the stroke is obtained. The radius of the link should not be less than eight times the travel, but must necessarily be adapted to the engine and varied according to the requirements of construction, assuming it in this case to be 42 inches. The length of the link is determined by the travel of the point *F*, which is ascertained from formula (11) which by the substitution of values becomes

$$b = \frac{13\sqrt{2.625^2 - 1.125^2}}{13 - 1.125} = 2.6 \text{ inches.}$$

Then the half length of the link, *Og*, is obtained by the substitution

of values in formula (12) when then becomes

$$Og = \frac{2.6}{0.41421} = 6.3 \text{ inches.}$$

With the proper allowance for the length of the link block the link should be at least 8½ inches long on each side of O or 17 inches in all. If the point K of the attachment of the eccentric rod is taken at 11½ inches from the link fulcrum, the radius of the eccentric crank, in order to give the link a throw of 45 degrees will be

$$11.5 \times \tan 22\tfrac{1}{2} \text{ deg.} = 11.5 \times 0.4142 = 4.75,$$

or by substitution of the values of formula (13)

$$b' = \frac{11.5 \times 2.6}{6.3} = 4.75,$$

so that it can be made 4¾ inches.

This is, however, subject to slight modifications depending on the angle between the motion center and the radius OK, but in ordinary cases this is so insignificant that it may be left out of consideration.

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MACHINERY'S REFERENCE SERIES

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No. 29—LOCOMOTIVE DESIGN

By Geo. L. Fowler and Carl J. Mellin

PART III.

SMOKEBOX, FRAMES, AND DRIVING MACHINERY

CONTENTS

CHAPTER I.

SMOKEBOX AND EXHAUST PIPE ARRANGEMENT. *

The arrangement of the smokebox and the draft appliances contained therein ranks among the more important of the details in the designing of a locomotive, for upon it largely depends the efficiency of the whole machine. Perhaps it is cn account of this very importance, coupled with the wide range of variables that must be taken into consideration, that there is as yet no standard construction that is acknowledged to be the one that will produce the highest efficiency under all conditions.

The problem to be solved, with some modifications, is to secure the largest possible exhaust nozzle, and uniform action on the fire over the whole surface of the bed.

The action of the exhaust is, for the most part, that of a jet of steam drawing air with it by the friction of its sides; though, at very low speeds, the plunger action is also influential since there is a perceptible interval of time between the exhausts. When the plunger action predominates, the form and diameter of the stack as well as the heights of the top and bottom of the same above the exhaust nozzles are matters of importance. The jet action is, however, the one that for the most part controls in all the working of the engine, but the smokebox details must be arranged to suit both. In considering the jet action, the length cf the stack, and, consequently, the height of its top above the nozzle, is of minor importance. It is evident that the shorter the stack the less will be the frictional resistance of the gases, so that it is useless to extend it beyond the point where they have obtained their maximum velocity. On the other hand, were the steam to be allowed to escape into the atmcsphere through too short a stack, the interval between exhausts would be sufficient to permit the air to rush back into the smokebox and firebox and, by destroying the partial vacuum that had been created, add very materially to the work that would have to be done. For this reason it is necessary to sharpen the exhaust by contracting the nozzle, thus prolonging the time of its action and increasing the velocity of the steam. On the other hand, if this contraction is made too great, there will be an excessive action

* The present number of MACHINERY'S Reference Series is the third part of a treatise on complete Locomotive Design, covered by Nos. 27, 28, 29 and 30 of the Series, and originally published in RAILWAY MACHINERY (the railway edition of MACHINERY). Each of the four parts of the complete work treats separately on one or more special features of locomotive design; and while the four parts make one homogeneous treatise on the whole subject, each part is complete by itself. In order to give concrete form to the examples and theoretical considerations, it is assumed that a consolidation freight locomotive and an Atlantic type passenger engine are being designed. It is further assumed that these locomotives are designed for a division 150 miles long, laid with rails weighing 75 pounds per yard, and with a ruling grade of one per cent ten miles in length.

and a breaking up of the bed of the fire, coupled with an undue back pressure in the cylinders.

The stack should be of such length that at moderate speeds one exhaust is entering at the bottom before the last of the preceding one has escaped at the top. At the same time it should be of such shape that resistances are reduced to a minimum. That this may be done it should increase in area in proportion to the loss of speed of the gases, by which means the inertia of those leading will be utilized to reduce the resistance of the succeeding ones; which, in turn, serves to increase the efficiency.

The exact amount of retardation of the gases in their passage from the nozzle to the air is only obtained by experiment, but it is evident that the stack should be flared, with the enlarged portion at the top. For practical purposes, however, the straight taper will answer every requirement with no noticeable difference from that of a form theoretically correct.

The most exhaustive experiments along this line that have thus far been made are probably those of Von Borries and Troske that were carried out in Germany in the early nineties and subsequently published in this country, from which it appears that the proper taper for a stack is approximately one in twelve and that its length should be three or four times the diameter at its smallest point or choke, a proposition that was confirmed by the committee of the American Railway Master Mechanics' Association on Exhaust Pipes and Steam Passages in 1896.

The results of the Von Borries and Troske experiments may be approximately expressed by the formulas:

$$d = 0.156 \sqrt{\frac{S - R}{S + 0.8\,R}} \qquad (1)$$

in which

d = diameter of the exhaust nozzle,

S = area through tubes,

R = grate area,

all expressed in inches.

$$h = 14\,d \qquad (2)$$

in which

h = height of the top of the stack above the top of the exhaust nozzle of a straight pipe.

$$D = 3.8\,d \qquad (3)$$

in which

D = diameter of the stack at the top.

$$D' = 0.65\,D \qquad (4)$$

in which

D' = an imaginary diameter which the bottom of the stack would have were it to be drawn down to the level of the top of the exhaust nozzle.

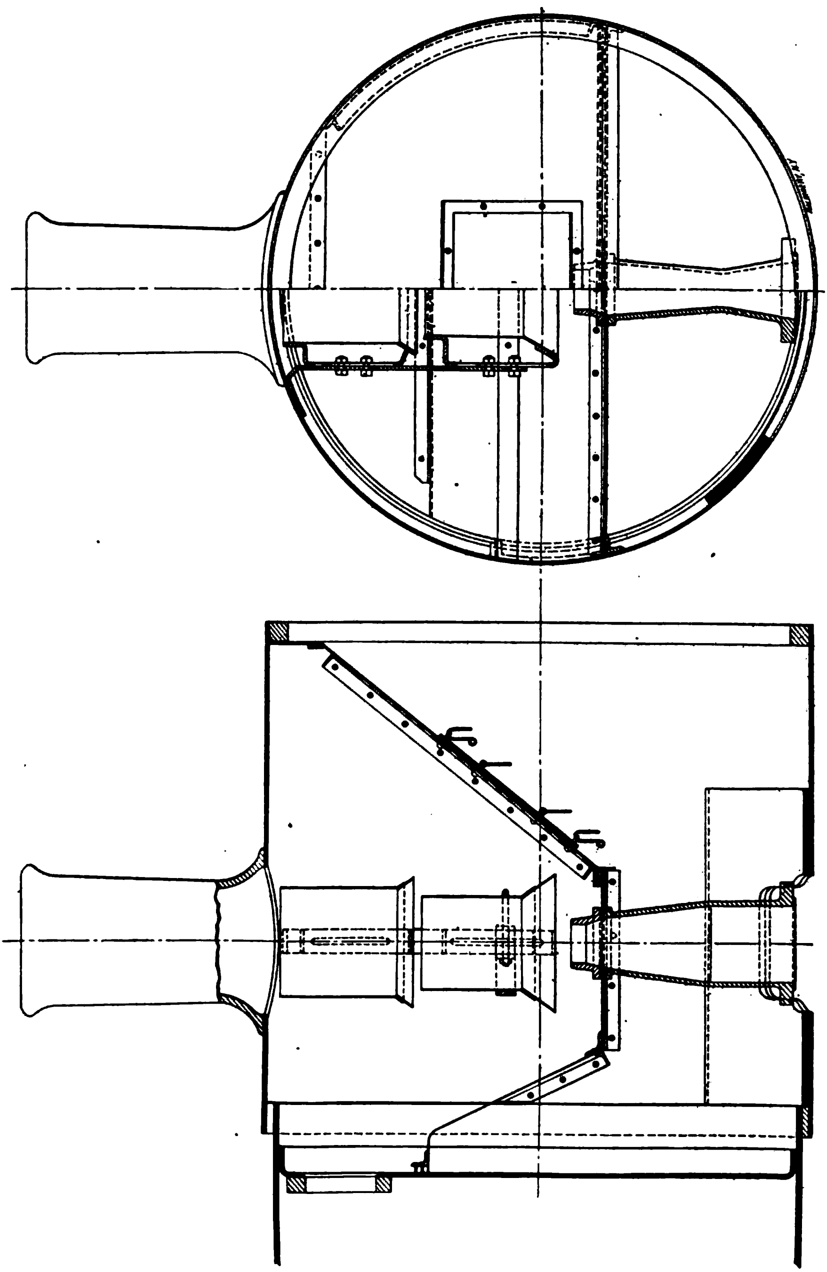

Fig. 1. Longitudinal and Cross-section of Smokebox of Express Passenger Locomotive.

The height C of the choke above the top of the nozzle should not be more than 0.4 h or

$$C = 0.4\,h \tag{5}$$

When circumstances do not permit the heights required by these formulas to be used, it is good practice to use at least the diameters of stack obtained by following the rules that have been laid down.

As for the effectiveness of the exhaust, it has been found that, under certain conditions, one pound of steam, discharged from the nozzle, will displace about 2.5 pounds of smokebox gases, as per the following formula, that has been deduced from earlier experiments.

$$A = D\sqrt{\frac{2\left(\frac{T}{S}\right)^2\left(1-\frac{V}{S}\right)}{\frac{l}{S}\left[4+2\left(\frac{T}{S}\right)^2\right]}} \tag{6}$$

in which

A = weight of gases displaced in pounds.
D = weight of steam discharged through the nozzle in pounds.
V = area of exhaust nozzle in sq. inches.
S = area of stack in sq. inches.
T = area through tubes in sq. inches.
l = a coefficient (approximately 4).

The weight of air displaced is in direct proportion to the weight of the steam discharged through the exhaust nozzle, when all adjustments have been properly made.

According to the experiments referred to above, the area of the stack, if straight, should be about twelve times the area of the exhaust nozzle, and in one that is well proportioned and tapered, the effect of the nozzle can be increased 12 per cent above that of a straight stack.

The base of the straight stack should be from 30 inches to 36 inches above the top of the nozzle, and the length about three times its own diameter. The choke or smallest diameter of a taper stack should be about 24 inches above the nozzle and its area should be about nine times the nozzle area. The length of the stack should be from three and a half to four times the diameter of the choke, with an area at the top not more than twice that of the choke.

These rules run remarkably close to each other, though the latter is considerably older than the former. They have, however, not yet secured the recognition in this country that they have abroad, and we find a variation of smokebox arrangements in use. This is due, in all probability, to the great variety of fuel burned.

Taking up the smokebox details in general, it will be found to be advantageous to locate the nozzle as low as the area across the box will allow. The baffle plate or diaphragm should be carried down, from above the top row of tubes, to the top of the nozzle, from which point it should be continued horizontally forward well to the front of the

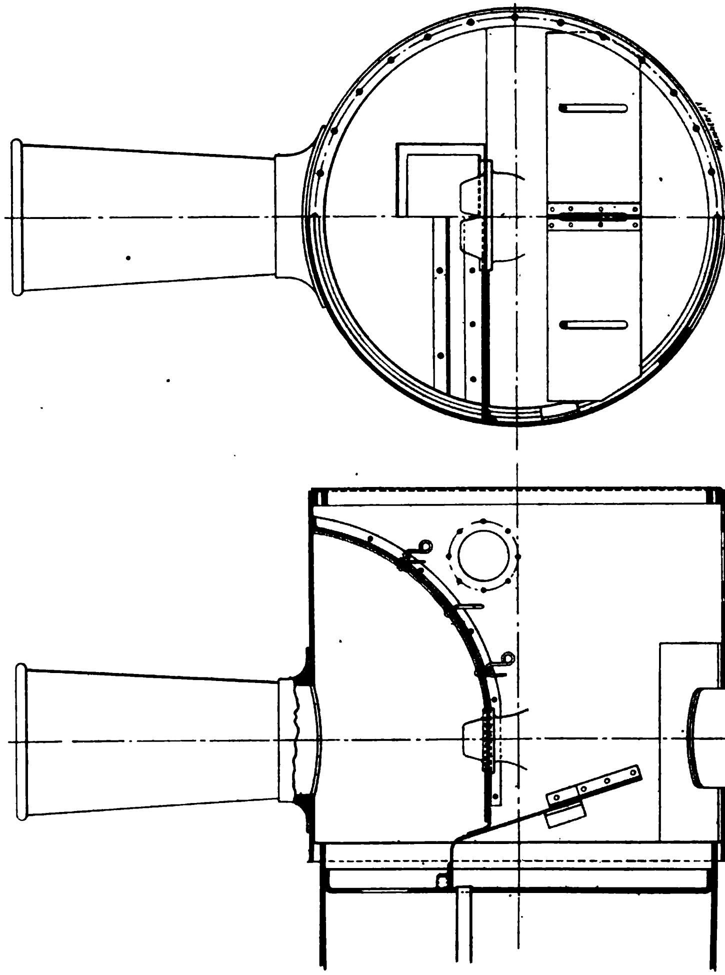

Fig. 2. Longitudinal and Cross-section of Smokebox of Consolidation Freight Locomotive.

exhaust pipe, from which the inclined spark-arresting netting starts and extends to the top of the smokebox as shown in Figs. 1 and 2. This general arrangement has come into common use, and, in many cases, constitutes the so-called self-cleaning front end.

In its entirety, it consists of an adjustable diaphragm plate so located as to allow but a limited passage for the gases and sparks between it and the bottom of the smokebox. The diaphragm is carried well forward to limit the area of the passage so that the cinders and gases are kept in state of constant circulation by the draft, with the result that the former are ground so small that they pass through the netting and out at the stack at a temperature below the igniting point, to which they have been cooled by the process.

The size of the smokebox is not a matter of much importance, except that a large one modifies the effect on the fire caused by the pulsations of the exhaust, on the same principle that an air chamber on a pump will cause a more uniform flow of water than would otherwise be obtained, whereas with a small box the fire will be more disturbed. On American locomotives the size of the smokebox has been reduced to a practical uniformity in the matter of length, ranging, as it does, from 6 feet to 7 feet, which is considered to be large enough to effect the desired uniformity in draft and affording sufficient room for the location of the diaphragms, netting and deflectors. Any length less than 4 feet would be considered small on a standard engine.

With the use of the low exhaust pipe, it is frequently found to be advantageous to introduce one or more draft or petticoat pipes between the exhaust nozzle and the bottom of the stack, as shown in Fig. 1, whereby, what amounts to several jets milder than that of the original issuing from the nozzle, are obtained. This holds especially for simple engines. For compounds it is advisable to extend the stack down into the smokebox to within 20 inches or 24 inches of the top of the nozzle and to secure the increased length of stack by which a more uniform action on the fire is obtained. In this arrangement of the smokebox no specific rules or proportions can be given for the working out of the details that will be of any value. The specific dimensions of the several parts will depend not only upon the size of the engine and the work that it is intended to do, but most particularly upon the fuel that is to be burned. For this a special adjustment is needed, and one that will give perfect satisfaction with a certain grade of coal will not be found suitable for another of a different character.

In Figs. 1 and 2 are given the longitudinal and cross sections of the smokeboxes for the two engines that are being developed, from which the general proportions and arrangement of the parts according to modern approved practice may be determined.

CHAPTER II.

THE FRAMES.

Although the frames are the foundation upon which the locomotive is constructed, they are not the first thing to be taken into consideration in the designing of the machine. As a support they can be varied in form to suit the requirements of the boiler, machinery and other parts and so only take on their final shape when these other parts have been arranged; though their attachments and presence must be borne in mind throughout the whole of the preparatory work. That they must receive careful consideration goes without saying, for upon them depends much in the matter of strength and rigidity, and perhaps even more in the way of maintenance, as broken frames are usually expensive to repair. In short, weak frames are a source of constant trouble in the way of failures and repairs, since they are frequently the cause of abnormal wear of the moving parts of the machine as well as a never ending, though sometimes obscure, agency in producing hot boxes.

As the width of the frame is necessarily limited, and, as the lateral stresses to which it is subjected are very great, it follows that it must not only be of ample size, but must be thoroughly braced so as to be able to withstand the shocks to which it is exposed when in service. It is impossible, however, to present any absolute formula by means of which the dimensions of a frame can be calculated. It is, of course, comparatively easy to calculate the stresses to which it will be subjected under the direct action of the steam in the cylinder, but this falls far short of being sufficient to provide for the diagonal and lateral stresses that are set up when the machine is worked heavily. These are of a very serious nature, and cannot be estimated when the speed is high or the track rough.

Experience shows that the lack of proper bracing is more often the cause of frame failures than the actual size of the sections used. For, if the frames are held rigidly in position both horizontally and vertically, these high running stresses become merely those of compression and tension. It is, therefore, apparent that the bracing or construction of the transverse frame work is a matter equal in importance to that of providing longitudinal strength. Hence, it cannot be too strongly enforced upon the attention of the designer that it is impossible to estimate the components of the forces that are set up in every direction, as in the case of the derailment of one or more pair of drivers. Nor can all of the varying conditions be ascertained as they exist at high speeds or on a rough track, where stresses of a momentary character are set up, that differ widely from second to second, and where the momentum of the engine is an important factor in determining the intensity of the side blows that are delivered.

As already stated the stresses imparted by the working of the engine are comparatively easy to estimate and this can be done by the following formula:

$$F = \frac{P \times \pi d^2 c}{4 e} \tag{7}$$

in which F = stress produced,

P = boiler pressure,

$\frac{\pi d^2}{4}$ = piston area,

c = distance from cylinder center to frame center on the opposite side of the engine in inches.

e = distance between frame centers in inches.

Practice has shown, however, that no matter what may have been the attempt to hold the frames in tram, it is impossible to eliminate the bending moments, and that an allowance must be made for them as well as for the other incidental stresses that have been referred to. The most practical way of accomplishing this is to assume a suitable fiber stress for that portion of the frame section which is above the pedestals and this may be done by the formula:

$$A = \frac{F}{S} = \frac{P \pi d^2 c}{4 e S}, \tag{8}$$

in which A = the section area of the frame,

F = total stress as obtained by formula (7),

S = fiber stress.

For the point in question the fiber stress should be placed at 4,000.

The same formula (8) holds good for the upper frame section between the pedestals as well as for the lower rail. The value of S, the fiber stress, should be changed to 5,000 for the former and 7,500 for the latter. These values give the lowest practical limit that it is advisable to use for wrought-iron frames. For cast steel frames, these values of S had better be made 3,000, 4,000 and 5,500 respectively. In all cases the nearest larger even dimensions should be used, so that the sectional area may not fall below the requirements of these formulas.

For practical reasons the width of the frame section should not vary more than ½ inch for engines having cylinders more than 18 inches in diameter, and the ratio of depth to width should be kept as near as possible to 5 to 4, the section above the pedestal being taken as a base. The greatest width of the pedestal legs should not be less than ½ inch. This, naturally, causes a deviation from the figures obtained by the formulas, and it is here that the necessity for good judgment comes in; for the effect of the reduction of sectional area by bolt holes must be considered, and this is particularly true in the case of small engines.

When larger holes than those used for the ordinary frame bolts are put in to take such parts as the equalizer or brake hanger bolts, an additional depth should be used at such places, to compensate for the material drilled out; and, in every instance, large fillets should be

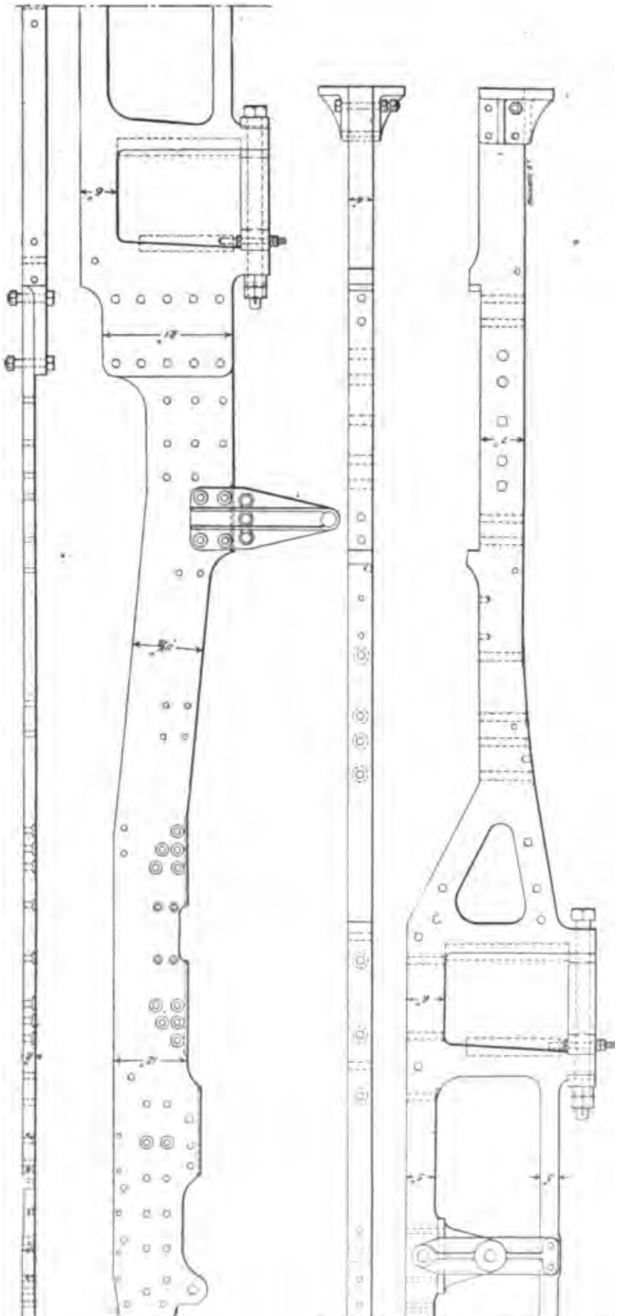

Fig. 3. Frame for Atlantic Type Passenger Locomotive.

used at all unions between the horizontal and vertical members of the frame.

The front extension that passes beneath the cylinders when a single bar frame is used, should be of an area equal to the sum of the areas of the top and bottom rails of the main frame. When top and bottom rails are used at the cylinders, the former should have the same sectional area as the top rail of the main frame and that of the latter should not be less than that above the pedestals. Finally the splices or connections between the front extension and the main frame should be made as long as possible, and be well bolted and keyed together.

At all points where there is any fitting at the corners of the frames for cylinders or braces and especially for the pedestal shoes and wedges, these corners should be well rounded not only on the frame, but on the piece to be fitted to the same. As a rule, the wedges are fitted to the back leg of the pedestal so that the greater pressure exerted on the shoe and the vertical front leg, in running ahead, may bear perpendicularly against the face of the same. The slope of the rear leg where the wedge has a bearing usually ranges from ¾ to 1 inch in 12, which gives ample opportunity to adjust for wear.

Reverting now to the bracing for the frame, it may be again reiterated that it can hardly be made too substantial. The cylinder castings which usually bear the greater portion of the burden of keeping the frames in line, should be relieved as far as possible by broad, well-ribbed cast steel crossties, having both horizontal and vertical extensions so as to form a rigid diagonal and transverse bracing between the upper and the lower rails, as well as by the horizontal and diagonal bracing that is needed between the frames themselves; and this bracing should be placed as close to the pedestal as possible.

No specific rule can be given either for the extent or the exact location of such bracing, because the requirements of the Stephenson link motion that is placed between the frames practically prevents the application of such bracing at points where it is most needed and where it would do the most good. The frequent results of this prohibition is to be found in a constant increase of frictional resistance and an ultimate breaking of the frames and cylinders. At the back end of the frames the footplate should extend along the frame for as great a distance as possible, as no adequate bracing can be put beneath the ashpan on the ordinary type cf engine.

The pedestal binders on heavy engines should be of wrought iron with deep notches to receive the lower ends of the legs. For small and medium-sized engines, however, a bolt and thimble makes a satisfactory binder. For many years wrought iron was the only material used in the frames of American locomotives, and the preceding discussion has been based on this practice, although the cast steel frame has also been considered. The latter possesses some decided advantages over the forged frame and it is now being extensively introduced. The chief of these advantages is the ability to use I-sections and otherwise so distribute the material that it is disposed to the best

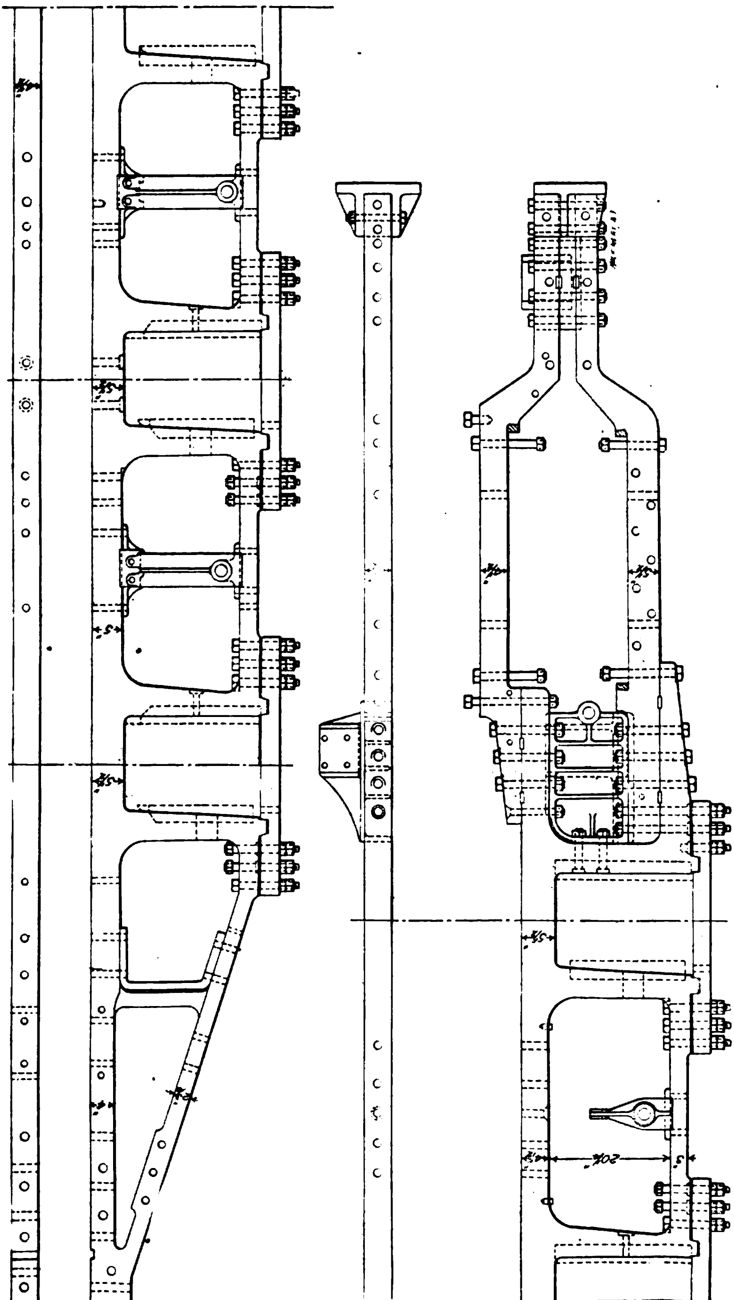

Fig. 4. Frame for Consolidation Freight Locomotive.

advantage to resist the stresses to which it will be subjected. This makes a stronger and a lighter structure, but has the resultant disadvantage in the difficulty which this lighter section offers to welding, if it is broken. This disadvantage more than offsets the primary advantage of the best disposition of the metal, and current practice is now leaning strongly toward the use of the rectangular section.

Another advantage in the use of the cast steel frame is to be found in the possibility of casting on all brackets and of making special arrangements for the attachment of auxiliary parts such as brake cylinders, braces, hangers, and the like; so that it is probable that cast steel frames of a section corresponding to that of those of wrought iron will be increasingly used.

Turning now to the frames of the two engines that we have under consideration let us see how their designs conform to the principles that have been thus far set forth. Fig. 3 shows the frame of the Atlantic type express locomotive, and Fig. 4 that for the consolidation locomotive.

In the case of the former an application of formula (7) becomes

$$F = \frac{200 \times 298.65 \times 64.5}{43} = 89{,}579 \text{ pounds.}$$

in which

$$\frac{\pi d^2}{4} = 298.65.$$

Then from formula (8) we have

$$A = \frac{89{,}579}{4{,}000} = 22.375 \text{ approximately.}$$

In this case 22.375 square inches is the lowest limit that would be allowable for the sectional area of the frame above the pedestal. In order to have even figures for the frame dimensions, the latter are given a depth of 6 inches and a width of 4 inches, producing a sectional area of 24 square inches. Likewise the upper frame section should, according to the formula and a fiber stress of 5,000 pounds, have an area of 18 square inches, which it has in the depth of 4.5 inches and width of 4 inches. The lower rail, with the fiber stress of 7,500 pounds should have an area of 11.9 square inches. It is made 12 by the depth of 3 inches and the width of 4 inches. In this case the single bar at the cylinder connection is made 7 inches deep instead of the 6 inches that is found over the pedestal, so as to compensate for the holes that are drilled there. In the consolidation locomotives the areas are less than those called for by the formula because of the distribution of the stresses through four pair of driving wheels instead of two. Large fillets are used at the union between vertical and horizontal members, and in the case of the frame of the consolidation engine the junction of the front and main sections is made very secure by means of bolts and wedges.

CHAPTER III.

CROSS-HEAD AND GUIDE BARS.

The piston rods and the pistons are at the genesis of the moving parts of the machinery, which must now be considered. Starting with the outer connection of the piston rod, the crosshead will be found to exist in various forms, from which a selection can be made to meet the requirements of any particular case. It should be borne in mind, however, that for heavy or high-speed engines the double-bar guide, or what is ordinarily known as the alligator type of guide and crosshead, will give the most satisfactory results, and is, consequently, the one that has been most extensively adopted. In fact, where conditions will admit of its being used, this type of crosshead will probably give the best satisfaction for any kind of an engine, even though it does necessitate the use of more metal by which the weight is greater than in a single-bar structure. In the designing of these parts it is also desirable to have the guides as close together as possible so as to reduce the distance to the piston rod to a minimum, and this holds for either a single or double-bar construction.

As for the crosshead pin, it might be very small so far as its power to resist the working stresses is concerned, since these are applied in shear only, but it must be made large enough so as to provide an ample bearing surface for the brasses in the main rod. In short, it should be of such diameter and length that the load does not exceed 5,000 pounds per square inch of projected area. The formula for these dimensions may be expressed as:

$$dl = \frac{P}{S} \qquad (9)$$

in which

d = diameter of crosshead pin,
l = length of crosshead pin,
P = total pressure upon the pin,
S = allowable pressure (5,000 pounds) per square inch of pin.

In this it is necessary to assume either d or l, and the pressure P may be calculated by multiplying the area of the piston by the boiler pressure. The length is the dimension most commonly assumed and this is taken to suit the width of the crosshead.

In connection with the designing of the pin it should be kept in mind that, because of the action of the engine on curves and the side play in the driving boxes, the bearing on the crosshead pin is liable to wear faster at the ends than in the center. Under such conditions there will be a bending moment in addition to the shear, so that it is advisable to calculate the stress to which it will be subjected under

full boiler pressure from the formula:

$$S = \frac{P\,l}{4\,m} \qquad (10)$$

in which

S = the fiber stress to which the metal will be subjected,
P = total piston pressure,
l = length of crosshead pin,
m = moment of resistance of the circular section of the pin.

The moment of resistance of circular sections may be found from the formula

$$\frac{\pi\,d^3}{32}$$

and for the ordinary diameters of crosshead pins ranging from 3 to 4½ inches the moments of resistance are as follows:

3	inches,	2.66	4	inches,	6.28
3¼	"	3.38	4¼	"	7.53
3½	"	4.21	4½	"	8.95
3¾	"	5.18			

An examination of these figures will show that they vary as the cubes of the diameters.

Owing to the great amount of motion between the guide and the crosshead and the exposure of the surfaces to external influence, the pressure at this point should not be allowed to exceed 70 pounds per square inch, while it is desirable to drop well below these figures if possible, even cutting it down to one-half that amount, which is done in many instances.

In this, as in other cases, it is well to remember that large bearing surfaces are a good investment. The pressure against the guide can be calculated from the formula:

$$P' = \frac{P\,r}{L} \qquad (11)$$

in which

P' = the total maximum pressure against the guide,
P = maximum pressure on the piston,
r = radius of the crank in inches,
L = length of connecting-rod in inches.

This is an approximate formula that gives results sufficiently accurate for steam engine practice where the ratio between the connecting-rod length and crank radius is not less than 6 to 1. The formula for thrust which gives the theoretical reaction on the guide is $P' = P \tan\theta$, in which P' = thrust against guide; P = maximum pressure on piston; θ = greatest angle made by connecting rod with axis of cylinder.

The area of the sliding surface of the crosshead will then be determined by

$$A = \frac{P}{t} \qquad (12)$$

where A = the required area of the sliding surface,
t = the allowable pressure per square inch.

According to the statement already made t should not be more than 70.

The strength of the guide should be such as to reduce the deflection to a minimum, which under no conditions should be allowed to exceed 1/32 inch and should be kept down to 0.01 inch or less when circumstances will allow. In cases where the cylinder center is raised above, but is still parallel with, a line drawn through the centers of the driving wheels, this distance should be added to the radius of the crank r in formula (11) as well as in the determination of the thickness of the guides, an allowance that will somewhat increase the value of the pressure against the guide, P', above what it would be were the crank radius alone used.

When the guides are forged the section is usually rectangular, in which case the moment of resistance is calculated from the well-known formula:

$$m = \frac{bh^2}{6} \tag{13}$$

in which
m = moment of resistance,
b = width of guide,
h = thickness of guide.

Having given the width, length and allowable fiber stress S, it is possible to calculate the thickness. The width is usually determined by the requirements of the construction, and the bars are fastened at both ends. This places the calculation on the basis,

$$\frac{P'l'}{4} = \frac{Sbh^2}{6}, \text{ or } h^2 = \frac{6\,P'l'}{4Sb}, \text{ and } h = \sqrt{\frac{3\,P'l'}{2Sb}} \tag{14}$$

in which l' = length of guide.

$P' = \dfrac{Pr}{L}$ of formula (11) with the necessary correction for the height of the center of the cylinders above the driving wheel centers.

It is always desirable and will usually be found to be necessary to check off this determination of the value of the thickness of the guides h, in order that the deflection may not exceed the amount given above. This deflection may be determined by the formula:

$$f = \frac{P'l'^3}{4Ebh^3} \tag{15}$$

in which
f = the deflection in inches,
$P' = \dfrac{Pr}{L}$, as before,
E = the modulus of elasticity, which for steel may be placed at 30,000,000, and b, h, and l' have the same values as in formulas (13) and (14).

The thickness of the guides having thus been determined, it is always well to add from ⅛ to ¼ inch to the amount so as to compensate for wear.

When the guide yoke is set ahead of the rear ends of the guides the length of the latter should be considered as the distance between the yoke bolts and those in the lugs of the cylinder head. The guide-yoke to which the back ends of the guides are fastened and by which they are supported must be strong enough to sustain the guide pressure P'. The special form that should be given to the yoke is largely dependent upon the other conditions of the design of the engine as a whole, and

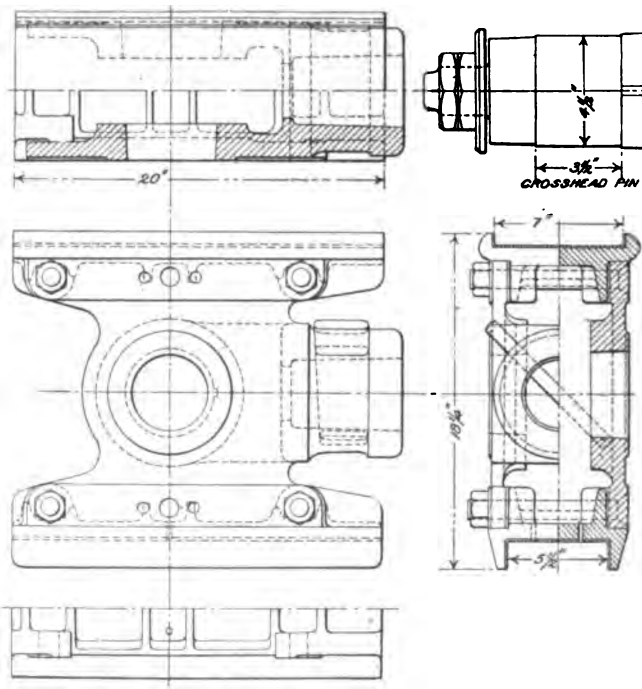

Fig. 5. Cross-head for Passenger Locomotive.

no definite shape can be recommended that would not call for wide variations. The points to be borne in mind in this connection are that the guide-yoke should not only be strong enough to carry the guides and hold them rigidly in position but should also be made to serve as one of the most valuable and efficient braces for the frame. Its position is one where a substantial support for the frame is needed, and it is customary to take advantage of the opportunity thus afforded and utilize it to the utmost.

With the principles thus enunciated it is possible to make a direct application to the engines that have been kept under consideration, always bearing in mind that exigencies of construction may require a

deviation, more or less pronounced, from the mathematical deductions, and that such a deviation is invariably made for the purpose of avoiding unusual or awkward dimensions.

Starting with the crosshead pin, by the substitution, in formula (9), of the values for the cylinder dimensions and steam pressures assumed to have been decided upon, and by taking a safe margin from the

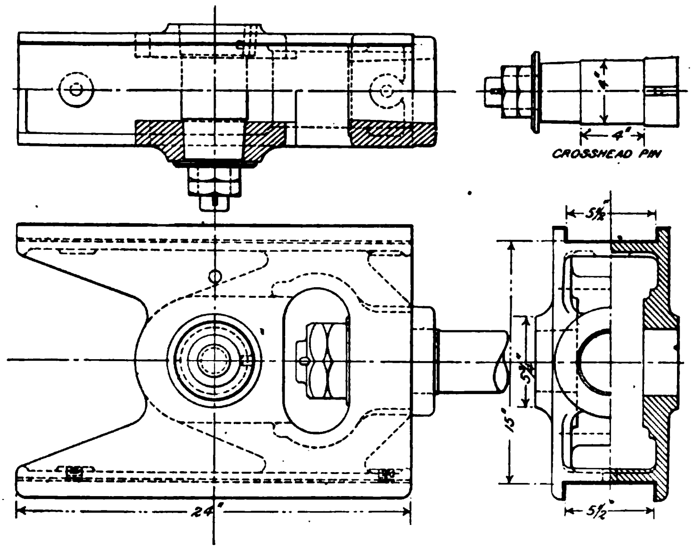

Fig. 6. Cross-head for Consolidation Freight Locomotive.

maximum allowable stress of 5,000 pounds, using 4,000 for the passenger engine, we have:

$$dl = \frac{59{,}730}{4{,}000} = 14.93,$$

while for the freight engine, where the speed is less, the 5,000 pounds may be retained, giving

$$dl = \frac{69{,}276}{5{,}000} = 13.85.$$

Figs. 5 and 6 illustrate the crossheads that have been designed for the passenger and freight engines respectively. In the case of the former, a pin 3½ inches long and 4.3 inches in diameter would satisfy the requirements; but in order to allow for some wear it has been made 4½ inches.

If these figures are checked by formula (10) we have

$$S = \frac{59{,}730 \times 3.5}{4 \times 8.95} = 5{,}840 \text{ pounds.}$$

This stress is much below what is required for strength in the pin, so that the size in this and other similar cases is governed by the

area of the bearing surface rather than by strength. Pursuing a similar course for the freight engine to whose crosshead a length of 4 inches is suited to the other exigencies of the design, it will be found that with a fiber stress allowance of 11,000 pounds a diameter of 4 inches will be required.

Passing to the area of wearing surface on the crosshead, and substituting the values that we have already obtained in formula (11) together with the length of the connecting rod, and noting that, in the case of the passenger locomotive, the crosshead center is 3 inches above the center of the driving wheels, while in the freight engine it is 2 inches above, we have:

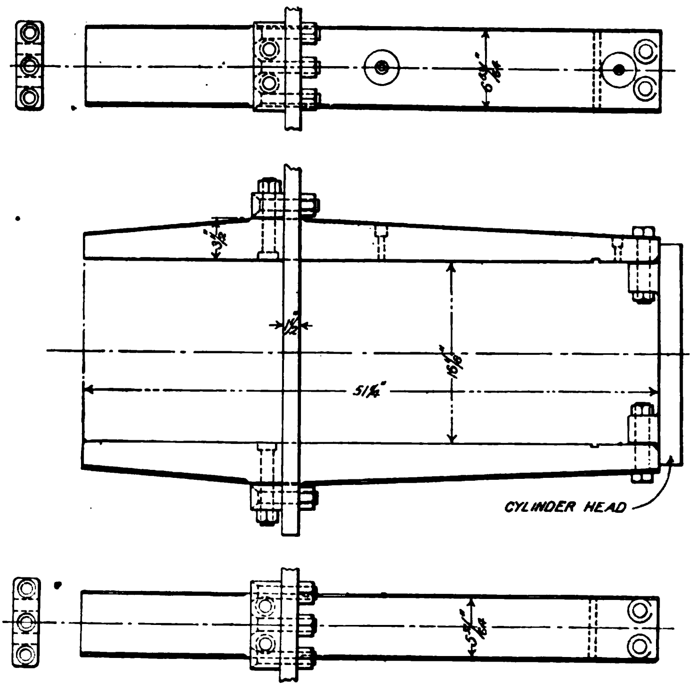

Fig. 7. Guide Bars For Passenger Locomotive.

$$P' = \frac{59{,}730 \times 16}{125} = 7{,}645 \text{ pounds for the passenger engine,}$$

and

$$P' = \frac{69{,}276 \times 15}{133.5} = 7{,}784 \text{ pounds for the consolidation engine.}$$

As there is an opportunity in these cases to use ample wearing surfaces, the crossheads are made 20 inches long and 7 inches wide for the passenger, and 24 inches long and 5.5 inches wide for the freight engine, dimensions which reduce the pressure per square inch to about 55 pounds and 67 pounds respectively.

The guides for the consolidation locomotive, illustrated in Fig. 8, have their width fixed by that of the bearing surface of the crosshead, while their length is determined by the length of the crosshead, the stroke and their own fastenings. With formula (14) a thickness would be obtained very much less than that used. The dimensions adopted are rendered necessary by the desirability of securing rigidity in accordance with formula (15), which then becomes

$$0.022 = \frac{7{,}784 \times (59\tfrac{3}{4})^3}{120{,}000{,}000 \times 7 \times h^3}; \quad h = 4.5 \text{ inches approx.}$$

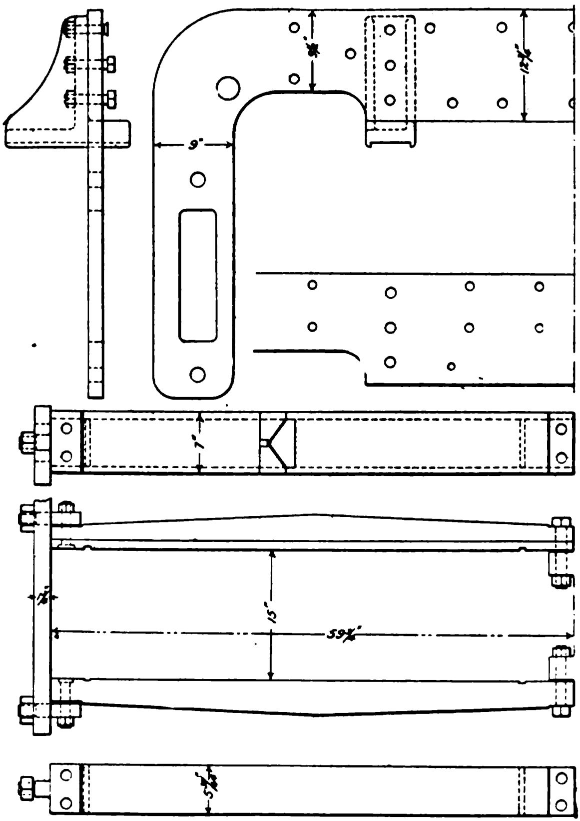

Fig. 8. Guides and Guide Yoke for Freight Locomotive.

In examining the engraving Fig. 8, it will be observed that the upper bar is widened above the lip of the crosshead in order to increase the strength, and at the same time reduce the thickness to a minimum, as well as for a protection against grit, by preventing dust and cinders working in between the crosshead and the top guide.

Two designs of guide yokes are shown for the two engines in Figs. 7 and 8. The one for the passenger engine, Fig. 7, holds the

guide near the center while the other holds it at the ends. Both are much heavier than the calculations for the mere static load would call for, to allow for incidental stresses due to various conditions such as derailments and other accidents that will put excessive stress on the yoke. In fact, the guide yoke must be designed in accordance with good judgment and past experience of what is suitable and not as the result of mathematical work. It should be always borne in mind that these formulas are to be used as outlines and approximations rather than for the final and unchangeable determination of the dimensions that are to be used.

Before leaving this subject attention may be called to a few details of current practice in crosshead construction. Steel castings have replaced cast iron and forgings, and when made from a steel casting the crosshead is usually made in one piece. This is the case on the consolidation locomotive, Fig. 6. For the passenger engine, however, it is built up with plates bolted in position. A decided advantage gained by the use of steel castings for this work is that it makes it possible to very materially lessen the weight of the crosshead. This is an important item that the designer should always remember, that the weight of the reciprocating parts should be kept down to the minimum, without a sacrifice of strength. In this, skill and judgment must be used and no invariable rule can be given for the disposition of the material where the stresses of operation must be cared for while every effort must be made to avoid the internal stresses that may be set up in the process of casting and cooling.

The ideal location for the crosshead pin is at the center of the length of the crosshead, a condition that is realized in the case of that of the consolidation engine. But where this is impossible, as in the case of the passenger engine, it should be placed as near the center as possible. Especial attention should also be paid to the boss around the seat of the piston rod to see that it is of ample strength and not liable to be cracked by keying.

These precautions must be borne in mind in the designing, else the result will be a broken crosshead or piston rod which never ends with a simple break of the first part yielding, but invariably involves other portions of the engine in a break-down that is crippling and costly, including, as it usually does, at least a cylinder head, if not the cylinder itself.

CHAPTER IV.

CONNECTING AND SIDE RODS.

The main or connecting-rod is the part of the mechanism by which the reciprocating motion of the crosshead is converted into the rotating motion of the crankpin. It is subjected not only to the tensile and compression stresses of the piston-rod but also to other stresses due to the vertical motion as well as to buckling loads imposed by the compression thrust on long columns. In short it is subjected to the stresses of compression, tension, horizontal deflection or bending due to compression, and vertical deflection due to compression, centrifugal force and inertia at high speeds. It is, therefore, of the greatest importance that the section should be as light as possible and yet of ample strength to carry the loads imposed. That is to say, it should be of such form and dimensions as to make the most economical utilization of the material, a consideration that is being more and more severely imposed with the increasing powers and speeds of locomotives. Both experience and mathematical calculations show that these requirements are best fulfilled by the I-section. In the matter of the material, it should be the best obtainable.

As to the area of the section, it should be large enough to withstand the crushing and pulling stresses to which it will be subjected; but, because of the importance of securing lightness of construction a comparatively high fiber stress can be allowed.

As the vertical bending moment is far greater than the horizontal, the section should be deep with broad top and bottom webs, in order that the material may be placed in the most favorable position for resisting these stresses. The vertical web must be of sufficient thickness not to buckle under the compressive load.

An analysis of the condition under which the rod works will show that, when the compression stresses have been cared for, there will be ample material to withstand the tensile loads. A further analysis will show that in compression the horizontal bending moment is simply that of a column with square ends while the vertical is that of one with round ends, and that the centrifugal inertia due to the motion acts as a load practically at right angles to the axis. The first consideration is for the crushing stress which is obtained from the formula:

$$S = \frac{P}{A}$$

in which

P = pressure on the piston,

A = area of the rod,

S = allowable pressure per square inch,
or where S is given the formula becomes

$$A = \frac{P}{S} \tag{16}$$

The same formula also holds for tension, and S may be taken to be the same, since the resistance of wrought iron and steel is about the same for both compression and tension.

The vertical bending moment is usually calculated on the basis of the piston pressure working at an estimated maximum speed in miles per hour equal to the number of inches in the diameter of the driving wheels. At such a high speed the piston pressure at mid-stroke will not be more than one-third of the maximum, so that the bending moment due to compression will be small, though that due to inertia will be maximum. The horizontal bending moment is based upon the full piston pressure, P, and is at its maximum at slow speeds.

The calculation of the various stresses to which the connecting-rod is subjected is a complicated matter, but must be carefully worked out in order to avoid not only weakness, but excess of weight; for on the one side, a weak rod is exceedingly dangerous and is apt to cause a disaster if it breaks at high speed when it is under the greatest stress due to the centrifugal inertia; while, on the other hand, too much material will set up other disturbing elements that affect the smooth running of the engine.

After calculating the direct tension and compressive stresses in accordance with the formula given, the next step is to determine the lateral stresses on the rod when considered as a column with square end supports for which the following formula can be employed:

$$F = \frac{B}{1 - \dfrac{N B l^2}{10 E R^2}} = \frac{10 B E R^2}{10 E R^2 - N B l^2} \tag{17}$$

in which

F = maximum compressive stress per square inch of concave side of column,

B = load per square inch of section of the rod $= \dfrac{P}{A}$,

E = modulus of elasticity = 30,000,000,

N = constant = ¼ for square bearing,

l = length of rod in inches,

R = radius of gyration of the section of the rod under consideration.

The same formula may be used for the vertical bending moment, except that as the rod has here become a column with rounded ends the value of $N = 1$ and the radius of gyration for the section must be taken about the horizontal axis. In both cases B is based on the full piston pressure.

The radius of gyration R is obtained by extracting the square root

of the quotient of the moment cf inertia I divided by the area of the section.

$$R = \sqrt{\frac{I}{A}} \tag{18}$$

in which

$$I = \frac{BH^3 - bh^3}{12}$$

where the several symbols of the second term of the equation are equal to the dimensions called for by the corresponding letters in Fig. 9.

When the speed is at the maximum it becomes necessary to combine the two bending forces due to compression and the inertia of centrifugal action respectively.

The general formula for centrifugal force is

$$C = \frac{Gv^2}{gr} \tag{19}$$

in which

C = centrifugal force,
G = weight of rod in pounds,
v = velocity in feet per second,
g = 32.2 = velocity acquired by gravity at the end of one second,
r = radius of motion in feet,

If the rate of revolution per minute is n then

$$v = \frac{2\,\pi\,r\,n}{60} = 0.1047\,r\,n$$

and

$$v^2 = 0.0109\,r^2\,n^2$$

whence

$$C = \frac{0.0109\,G\,r^2\,n^2}{32.2\,r} = 0.00034Grn^2 \tag{20}$$

Then taking the assumed maximum speed in miles per hour, V, as equal to the diameter, D, of the driving wheels in inches we have

$$n = \frac{V \times 5280 \times 12}{D \times \pi \times 60} = \frac{336\,V}{D} = 336$$

whence $n^2 = 112{,}896$.

The formula (20) then becomes

$$C = 38.38\,Gr \text{ or } 38.4\,Gr.$$

For convenience this may be converted into terms of the stroke, s, of the piston in inches.

$$s = 2r \times 12 = 24r$$

whence

$$C = \frac{38.4\,G\,s}{24} = 1.6Gs. \tag{21}$$

This is the simplest form in which the centrifugal force can be expressed and is as applicable to the side as to the main rods, though

the effects will differ with the difference in the motion of the two rods.

Taking the main rod first, the centrifugal force is at zero at the crosshead and at the maximum at the crankpin. If the centrifugal motion were to be considered as though the whole rod were in cirular motion, as in the case of the side-rod, with the length of the rod $= l$ and the centrifugal force $= C$, then the load can be represented by a rectangle as indicated by the dotted lines in Fig. 10, in which case the rod would be supposed to be loaded uniformly throughout its whole length with a burden equal to the centrifugal force. But as there is no load at the crosshead end, the rectangle may be divided diagonally, forming a right-angled triangle whose apex is above the crankpin and which may be taken to represent the centrifugal force as applied. The center of gravity of this triangle is at a distance of $l/3$ from the crank-

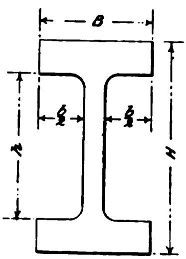

Fig. 9.

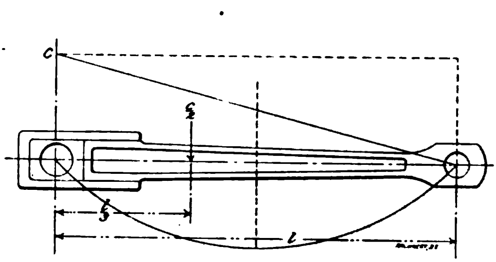

Fig. 10.

pin, and it is at this point that the whole of the centrifugal force ($C/2$), as represented by the triangle, may be considered to be applied to the rod.

As, however, the main rod is heavier at the crankpin than at the crosshead end, this is not strictly true because the load would fall somewhat nearer the crankpin than $l/3$, but the maximum stress on the rod will fall between 1/3 and 1/2 the length of the rod from the crankpin. The exact determination of this point would involve complicated calculations of no real value, in that a safety margin must be provided in any event, and the movement of the point of load application nearer the center merely decreases the margin.

It is, therefore, safer and better to assume the inertia load of $C/2$ to be at the middle of the rod or on the same section as the maximum vertical bending moment due to compression, working as indicated by the curved line in Fig. 10. As the rod is supported at both ends, the maximum moment (M) due to centrifugal action will then obtain when

$$M = \frac{C\,l}{2 \times 8} = \frac{1.6\,G\,s\,l}{16} = 0.1\,G\,s\,l \qquad (22)$$

The fiber stress (T) of the rod at this point due to inertia will then be

$$T = \frac{0.1\,G\,s\,l}{W} \qquad (23)$$

where W = modulus of section.

The combined vertical bending stress will then be equal to the sum of that obtained by formulas (17) and (23)

$$K = \frac{10\,B\,E\,R^2}{10\,E\,R^2 - N\,B\,l^2} + \frac{0.1\,G\,s\,l}{W} \qquad (24)$$

When the speed has reached the indicated maximum the value of B falls considerably below the full boiler pressure, because at midstroke the steam in the cylinder is, as already stated, rarely more than one-third the initial pressure. But as a further precaution in the case of temporary spurts of speed, as in the slipping of the wheels, B may be taken as equal to $\frac{P}{2A}$

The meaning of the symbols used in formula (24) is the same as that previously indicated for formulas (17), (19), (21), and (23).

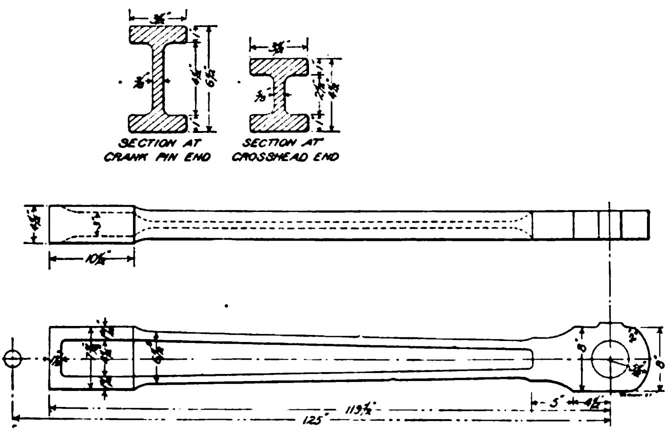

Fig. 11. Main Rod for Passenger Locomotive.

The vertical fiber stress K to which the rod will be subjected in service having thus been determined, the dimensions and proper section to be used can only be ascertained by the assumption of some definite size and then applying the formula to determine its strength. If the section is found to be too light or too heavy, it must be increased or decreased accordingly until the proper figures are found that will meet the requirements of the case in hand.

The stresses at the several points of the connecting-rod can be shown graphically by laying out a momentum curve on the lower side of the rod and an inertia triangle above it as shown in Fig. 10.

In the case of the side rod, the inertia effect due to centrifugal action is represented correctly by the rectangle of Fig. 10, which is equivalent to a load uniformly distributed over its whole length. As for the compression stresses due to the thrust of the piston, these are prob-

ably greater at low than at high speeds. The strain on the side rod must always be that of overcoming the slip of the coupled wheels to which it is connected, due to the slight inequalities of circumference that always exist, and the resistance of its own inertia stresses.

Turning now to the application of the principles that have been laid down to the rods of the engines whose parts have been used as a basis of comparison, the main-rod of the passenger engine is shown in Fig. 11, the side-rod for the same in Fig. 12. The main and side-rods of the consolidation freight locomotive will be shown later.

In the case of the passenger locomotive the diameter of the cylinder is 19½ inches, and the boiler pressure 200 pounds per square inch. This makes the pressure per square inch of area $B = \frac{59{,}730}{A}$ pounds, and for the consolidation locomotive with 21-inch cylinders $\frac{69{,}276}{A}$ pounds.

The diameter of the driving wheels (D) of the two engines are 72

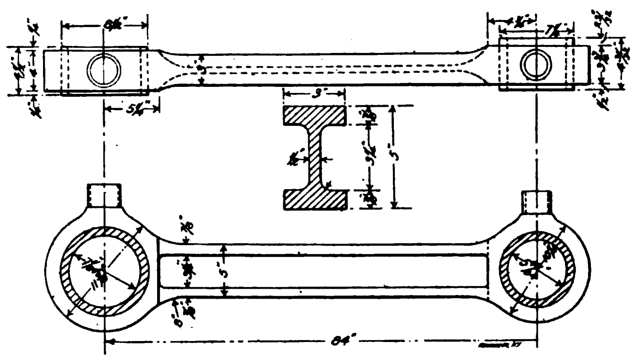

Fig. 12. Side Rod for Passenger Locomotive.

inches and 57 inches respectively so that the maximum speeds for which the rod stresses should be calculated are 72 miles per hour for the Atlantic and 57 miles per hour for the consolidation engine.

The first thing to be done is to determine the minimum area of the section of the rod according to formula (16) in which it is necessary to assume the value of S which may be placed at 7,000 as an approximation to meet requirements for the other stresses. For the passenger locomotive the formula then becomes:

$$A = \frac{59{,}730}{7{,}000} = 8.53 \text{ square inches.}$$

As already stated, the method of determining the values of the various factors in formula (24) is to assume a given section and weight of rod. When the radius of gyration and section modulus of this section has been substituted in the formula, the value of K, as so determined, must not exceed the allowable stresses that are settled empirically.

If, then, we take the main rod of the Atlantic locomotive as shown in Fig. 11, the weight will be found to be 404 pounds.

From formula (18) the radius of gyration R will be found to be equal to 2.03 and the section modulus

$$W = \frac{I}{q} = \frac{38.2539}{2.75} = 13.9$$

in which $q =$ the distance from the neutral axis to the outer fibers.

By the substitution of these values formula (24) becomes

$$K = \frac{10 \times 3{,}500 \times 30{,}000{,}000 \times 4}{10 \times 30{,}000{,}000 \times 4 - 1 \times 3{,}500 \times 15{,}625} + \frac{1 \times 400 \times 24 \times 125}{14}$$
$= 12{,}239.$

$K = F + T = 3{,}667 + 8{,}572 = 12{,}239$ in the above demonstration, or

$$F = \frac{10 \times 3{,}500 \times 30{,}000{,}000 \times 2^2}{10 \times 30{,}000{,}000 \times 2^2 - 1 \times 3{,}500 \times 125^2} = 3{,}667 \text{ pounds per square}$$

inch for vertical compressing bending stress under high speed, where

$$B = \frac{P}{2A} = \frac{60{,}000}{2 \times 8.5} = 3{,}500 \text{ pounds.}$$

$$T = \frac{0.1 \times 400 \times 24 \times 125}{14} = 8{,}572.$$

This falls within the stresses allowable for steel in this position which should not exceed 14,000 pounds fiber stress, and should be held as much below this as possible. In this calculation the area and section at the center of the rod are used as the basis of the work. The formulas can be applied in the same way to the side-rods as well as to all the rods of the consolidation locomotive. The details of the construction of these rods will be discussed later.

Side Rods.

It will be seen by a reference to Fig. 12, which represents the side rod of a passenger locomotive having but two pairs of driving wheels, that the construction is exceedingly simple and that all of the keys and gibs that formerly constituted so unimportant a part of the rods of engines have been entirely dispensed with. The rod in the present construction consists merely of a fluted bar with solid ends into which brass bushings are pressed. These are made to fit over the crank pins with an easy play and afford no means of adjustment to take up the wear. Such rods are the present universal practice on American locomotives. They are used until they become so worn that the pound on the pins is objectionable, when the bushings are renewed.

The side rods of a consolidation locomotive as well as those used on engines having more than two pairs of driving wheels, such as the ten-wheel and mogul classes, require a special arrangement of side rods. It is evident that the moving of a locomotive over a rough track involves a variation in the height of the driving wheels so that a rigid rod extending the full length of the wheel base would be impossible to

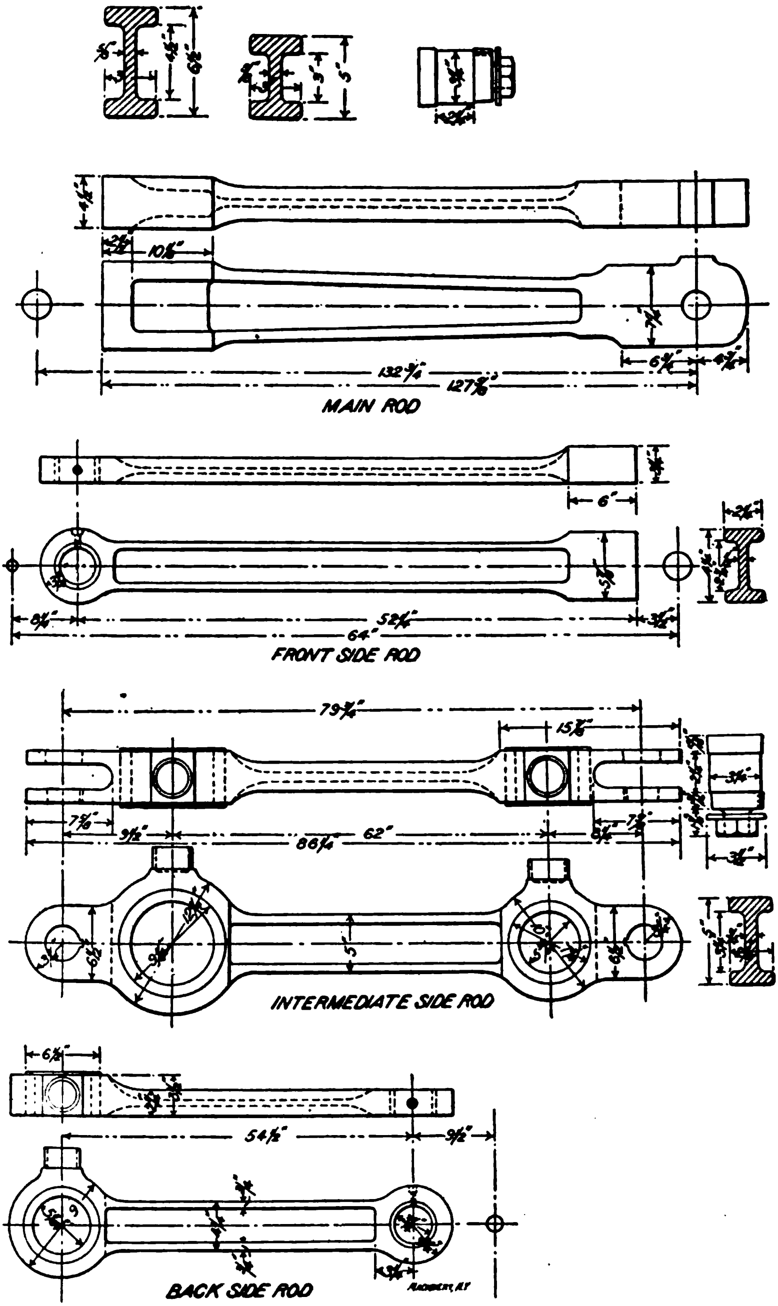

Fig 18. Main and Side Rods for Consolidation Locomotive

operate safely. This necessitates the use of a horizontal joint at each wheel so that there may be a free vertical movement between the wheels without causing any cramping cf the rod.

The general form of these rods is the same as that shown for the

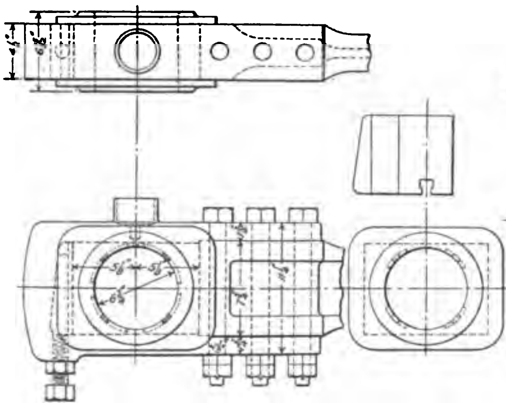

Fig. 14. Main Crank-Pin Connection for Connecting-Rod.

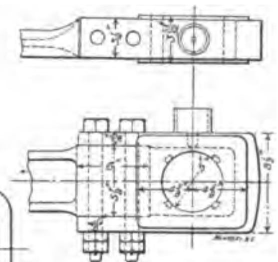

Fig. 15. Front End of Front Section of Side Rod.

passenger locomotive with four driving wheels. That is to say, the body of the rod is fluted and the bearings are solid brasses that admit of no adjustment; but in the case of the consolidation locomotive that we have in view, the side rod consists of three sections. The principal

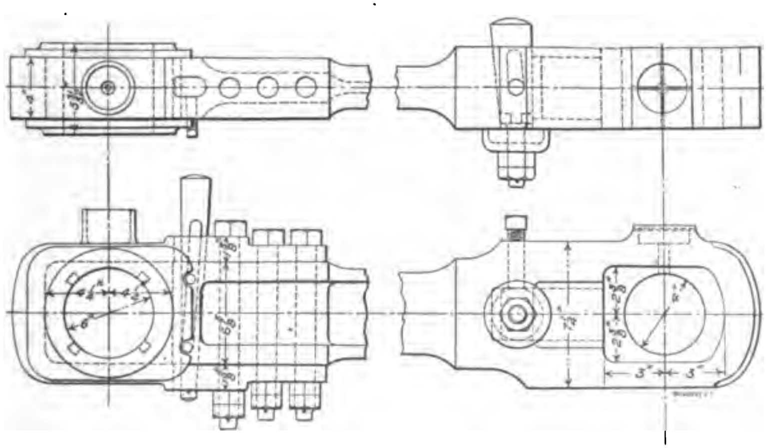

Fig. 16. Stub Ends of Connecting-Rod of Atlantic Locomotive.

rod reaching from the main crank pin to the pin in the third pair of wheels is solid like the passenger engine rod; and, as far as its action in connection with these two pairs of wheels is concerned, is the same. Each end of this rod carries a prolongation forming a knuckle joint with the stud ends attached to the front and back sections of the rod.

The pin for these places has a tapered head at the back to fit into a counterbore in the back lip in order to clear the face of the wheel. The rear section of the rod is also solid as far as the connection between the two rear pairs of drivers is concerned and is pivoted in the fork of the intermediate section. Thus as far as adjustment is concerned there is none possible longitudinally between the three rear pairs of wheels of this consolidation locomotive.

This is as far as it has been found to be advisable to carry this scheme of non-adjustment. Hence the front section of the side rod is provided with a stub and strap at its front end, by which the distance between the centers of the front and back crank-pin brasses can be adjusted. Even here, however, the adjustment is made with liners and not with keys and wedges. Fig. 13 shows the general construction of the main and side rods on the consolidation locomotive. Fig. 15 shows the arrangement of the details of the stub end of the front end of the front section of the side rod. The brass is shown with babbitt pockets, and is in a strap to which the oil cup is forged solid and held by two bolts of 1⅛ inch diameter. Any adjustments as to length of this rod necessitates the removal of the strap, the filing of the brass and the insertion of liners. Fig. 14 shows the arrangements of the details of the stub end that is used for the main crank-pin connection of the connecting rod. In this the strap is held to the end of the rod by three bolts of 1⅜ inch diameter each, and an adjustment is provided at the outer end by a wedge acting against the brass, which can be moved up and down by a screw bolt. Of course no adjustment can be made here without taking the brass out and filing it. There are many forms of stub ends that vary in detail from those shown and which are usually based on a matter of choice in construction rather than any inherent merits of the one over the other, the general principles of all being about as stated here.

CHAPTER V.

CRANK-PINS AND AXLES.

Turning to the subject of the crank-pin the two considerations that are most prominent are the bending stresses to which it is subjected and the necessity for the provision of a proper bearing surface. The latter is usually the one that determines the size of the pin; but whether it be the limitation of the fiber stress due to bending or the necessity of providing an ample bearing surface, the one calling for the larger diameter of pin is the one that should be selected to control. As for the distribution of the work to the several pins, it is a problem whose answer has not yet been reached, as the loads vary constantly, so that it is impossible to lay down any hard and fast rule in regard to it. If the bearings were perfect and without any play, the pins might have a truly proportional load to carry; but, as this is never the case, the best that can be done is to agree upon an arbitrary division. There are moments, for instance, when the main crank-pin will have to sustain the whole load, as when it is on the dead center and the side-rod bearings have considerable play. In other positions of the crank, the side rod is counteracting the stress on the main pin, so that if this assistance from the side-rod is considered, a comparatively high fiber stress can be allowed. This pin should have a wheel fit somewhat larger than the bearing. The length of the leverage of the main rod on the pin is usually taken from the wheel hub to the center of the bearing, and the diameter at the former point, that is, just outside the wheel hub, as shown in Fig. 18, may be found by the formula

$$D = \sqrt[3]{\frac{P l \times 32}{\pi S}} = \sqrt[3]{\frac{10.2 P l}{S}} \qquad (25)$$

In which

D = diameter of pin just outside the wheel fit,
P = the total pressure exerted on the pin by the piston,
S = the allowable fiber stress for the material,
l = the length of the pin as shown in Fig. 18.

The fiber stress should be limited to 16,000 pounds per square inch for steel and 14,000 pounds for wrought iron.

There should be only a small reduction of diameter of the pin for the bearing so that the requisite surface can be obtained without making the pin too long. This usually calls for so large a diameter that calculation of strength at the shoulders is unnecessary. The required projected area for the bearings must be such that the pressure per square inch does not exceed 1,600 pounds per square inch, or

$$\frac{P}{D\,l'} = \text{or} < 1{,}600 \qquad (26)$$

in which the letters have the same meaning as in the case of formula (25), except that l' = length of bearing on the pin.

The side rod crank-pins will never have any greater pressure brought upon them than that needed to slip the pair of wheels to which they belong so that the general principle is established that the maximum stress to which a side-rod pin can be subjected is that required to slip

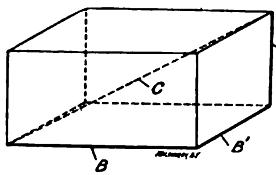

Fig. 17. Crank-Pin Stress Diagram.

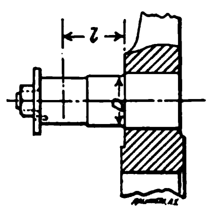

Fig. 18. Dimensions for Formula (25).

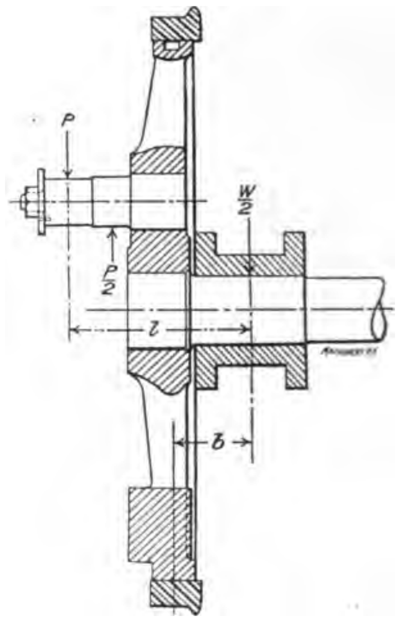

Fig. 19. Dimensions for Formulas (28) to (32).

the wheels to which it belongs. This may be calculated by the formula

$$P' = \frac{0.3\,W'\,D}{g} \qquad (27)$$

in which

P' = the frictional resistance of the driving wheels against the rails, transferred to the crank-pin,

W' = weight of the wheel with its load on the rails,

D = diameter of the driving wheels, in inches,

g = stroke of the piston, in inches.

In order to find the diameter of the pin, the value of $P' \times l'$ is substituted in formula (25) for $P \times l$. In this the fiber stress should be placed at 12,000 pounds for a side-rod pin instead of the 16,000 pounds used with the main pin, because certain reasons, on account of which it was allowable there, do not exist here in the case of the side-rod pins. Having found the diameter required to meet the required fiber

stress, the bearing surface may be proportioned according to formula (26) substituting, however, the value of P' as given in formula (27). It will be observed that the values of P and P' as given in these three formulas indicate the highest stress that can be applied to the pins, so that the working pressure of 1,600 pounds per square inch can only be exerted at very low speeds when the full boiler pressure is put upon the piston.

It is quite apparent, without argument, that the driving axles are among the most important of the details of a locomotive. Like the main rod, they are subjected to many and varying forces, such as the horizontal and vertical bending moments and torsion. The vertical bending moment, when considered as static, is the smallest to which the axle is subjected and is readily calculated. On the other hand, the shocks to which the part is exposed at high speeds are almost impossible to determine. At the same time the torsional stress under those same conditions, is comparatively slight, so that the increase of one may be considered to compensate for the decrease in the other. The horizontal bending force is due directly to the pressure exerted by the piston. The same is true of the torsional stress, whereas the vertical bending moment is due to the weight on the axle.

The torsional stress is never in excess of P' in formula (27) since anything above this is transferred through the side rods to the other drivers, as one crank is in a favorable position to slip the wheels, when the other is on the dead center, thus taking up all of the slack in the rods, and relieving the axle from the full bending force of the piston. It is, therefore, customary to consider this bending moment as equal to one-half that exerted by the piston or $\frac{Pl}{2}$, and the formula for the horizontal bending moment alone would be

$$B = \sqrt[3]{\frac{32\,Pl}{2\,\pi\,S}} = \sqrt[3]{\frac{5.1\,Pl}{S}} \qquad (28)$$

in which

B = the required diameter of an axle that would resist the horizontal bending moment,

P = the piston pressure,

l = distance between the center of the main rod and the center of the journal box,

S = allowable fiber stress.

The weight on the axle is a constant load and must, therefore, always be taken into consideration in connection with the horizontal bending moment, whereby the resulting bending moment of these two forces is equal to the hypothenuse of a right-angled triangle of which they are, themselves, the two sides. Consider, then, the total load on the axle as equal to W and the horizontal distance between the center of the rail and the center of the driving box as b, as indicated in Fig. 19. The vertical bending moment for $\frac{W}{2}$ or that for each box

will be obtained when

$$B' = \sqrt[3]{\frac{32\, W\, b}{2\,\pi\, S}} = \sqrt[3]{\frac{5.1\, W b}{S}} \qquad (29)$$

in which

B' = the vertical bending moment.

When the crank stands on the quarter the torsional stress is to be added and may be expressed in the same way as the bending forces. If this stress of torsion is indicated as B'', we then have

$$B'' = \sqrt[3]{\frac{16\, t\, r}{2\,\pi\, S}} = \sqrt[3]{\frac{5.1\, t\, r}{2\, S}} \qquad (30)$$

in which

t = the resisting force at the crank pin to the turning or slipping of the wheels,

r = radius of crank, in inches.

This value t is to be found in the same manner as that of P' in formula (27).

Thus

$$t = \frac{0.3\, W'\, D}{g} \qquad (31)$$

in which

W' = total load of one pair of wheels on the rail,

D = diameter of drivers in inches,

g = stroke of piston in inches,

0.3 = coefficient of friction.

It will be noted, of course, that all of the power required to turn or slip a pair of wheels does not go through the axle. One-half is absorbed directly by the wheel to which the crank is applied, and the other half, or only enough to turn the wheel at the opposite end of the axle, goes through the latter, thus reducing the torsion by one-half, a condition that is provided for in formula (30) by the constant 2 in the denominator of the general formula for torsion.

By combining formulas (28), (29), and (30), it is possible to determine the required diameter D of the axle, thus:

$$D = \sqrt[3]{\sqrt{\left(\frac{5.1\, P l}{S}\right)^2 + \left(\frac{5.1\, W b}{S}\right)^2 + \left(\frac{5.1\, t\, r}{2\, S}\right)^2}}$$

This reduces to the form

$$D = \sqrt[3]{\frac{5.1}{S}\sqrt{(P\, l)^2 + (W\, b)^2 + \left(\frac{t\, r}{2}\right)^2}} \qquad (32)$$

which is preferable as decreasing the size of the number to be squared.

The whole problem can be indicated graphically by measuring the diagonal C of a parallelopiped, where the value of B obtained in formula (28) is the length; that of B' in formula (29) the width, and that of B'' of formula (30) the height, as shown in Fig. 17.

In applying formula (32) to the determination of the diameter of the

axle for the consolidation locomotive, the following values for the several symbols will be obtained.

$P = 69{,}270$, or in round numbers 70,000 pounds,

$l = 22$ inches,

$W = 40{,}000$ pounds less the weight of the wheels and axles or 32,000 pounds,

$b = 10$ inches = horizontal distance from center of rail to center of box,

$$t = \frac{0.3\ W'\ d}{g} = \frac{0.3 \times 40{,}000 \times 57}{26} = 26{,}000 \text{ pounds,}$$

r = radius of crank = 13 inches,

S = allowable fiber stress = 16,000 pounds.

With these values the diameter of the axle (D) becomes

$$D = \sqrt[3]{\frac{5.1}{16{,}000}\sqrt{(70{,}000 \times 22)^2 + (32{,}000 \times 10)^2 + \left(\frac{26{,}000 \times 13}{2}\right)^2}}$$

This may be simplified by cancellation if written

$$D = \sqrt[3]{\sqrt{\left(\frac{5.1 \times 70{,}000 \times 22}{16{,}000}\right)^2 + \left(\frac{5.1 \times 32{,}000 \times 10}{16{,}000}\right)^2 + \left(\frac{5.1 \times 26{,}000 \times 13}{2 \times 16{,}000}\right)^2}}.$$

and

$$D = \sqrt[3]{\sqrt{\left(\frac{5.1 \times 70 \times 11}{8}\right)^2 + (5.1 \times 2 \times 10)^2 + \left(\frac{5.1 \times 13 \times 13}{16}\right)^2}}$$

$$D = \sqrt[3]{\sqrt{(491)^2 + (102)^2 + (53.8)^2}} = \sqrt[3]{\sqrt{241{,}081 + 10{,}404 + 2{,}894}}$$

$$D = \sqrt[3]{504} = 7.96 \text{ inches.}$$

By making a suitable allowance for wear and truing, the axle is made 9 inches in diameter at the journal.

The front and rear axles or all except the main driving axle may, of course, be made smaller, but it is customary to make them all of the same diameter, though exceptions to this rule are frequent. The same formulas may be applied to their determinations, by the substitution of the proper values for the several symbols. Attention should be called to the fact that the horizontal bending moment, which is the greatest in the case of the main axle, is reduced to less than one third on the other axles of a consolidation locomotive.

Turning now to the crankpin and applying formulas (25), (26), and (27) to the consolidation locomotive, the bending forces of the main pin at the wheel seat will give a diameter

$$D = \sqrt[3]{\frac{10.2\,Pl}{S}} = \sqrt[3]{\frac{10.2 \times 70{,}000 \times 8.375}{16{,}000}} = \sqrt[3]{373} = 7.19 \text{ inches.} \quad (25)$$

The diameter of this crankpin is made 7¼ inches, which is quite

sufficient when the liberal allowance made in the formula for the pressure on the piston is considered, as this is rarely reached, and the statement is emphasized by the fact that no allowance has been made for the reaction of the side rod when the wheels are slipped into bearing by the opposite crank which is on the quarter at the time the maximum stress comes on the one at the dead center where the calculation is based.

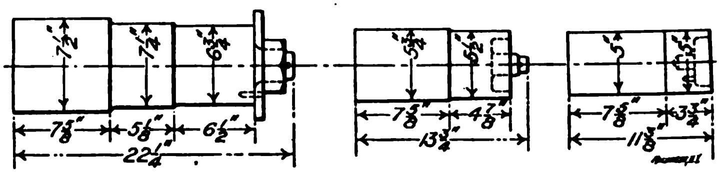

Fig. 20. Crank-pins used on Consolidation Locomotives.

This diameter will be that of the side rod bearing which, because of its large diameter, can be made very short. The pressure will seldom be more than three-quarters that of the main pin, and usually will be less. However, it will be found that, in the case of this consolidation locomotive the bearings are made longer than the 1,600 pounds pressure per square inch required, for constructive reasons. Thus

$$l = \frac{3 \times 70{,}000}{4 \times 7.25 \times 1{,}600} = 4.5 \text{ inches approx.}$$

But for the reasons just given it is made 5⅛ inches long.

The pins are usually reduced in diameter for the main bearing, in

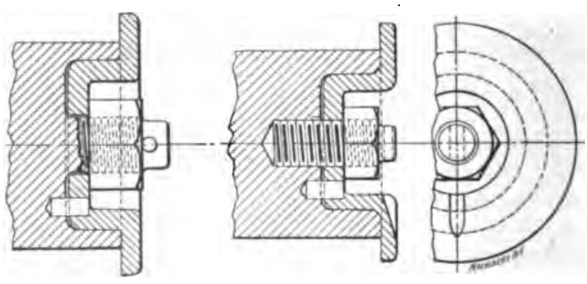

Fig. 21. Method of Attaching Crank-pin Collar.

order that the connections may be as light as possible. In the present case the projected area called for amounts to

$$D\,l' = \frac{P}{1{,}600} = \frac{70{,}000}{1{,}600} = 43.75 \text{ or a bearing about } 6.25 \times 7 \text{ inches.} \quad (26)$$

Taking formula (27) for the side-rod pins we have

$$P' = \frac{0.3\,W'D}{g} = \frac{0.3 \times 40{,}000 \times 57}{26} = \text{about } 26{,}000 \text{ pounds}$$

and

$$\frac{26,000}{1,600} = 16.2 \text{ square inches of bearing area}$$

and

$$D = \sqrt[3]{\frac{10.2\,Pl}{S}} = \sqrt[3]{\frac{10.2 \times 40,000 \times 2}{16,000}} = \sqrt[3]{51} = 3.7 \text{ inches, or a}$$

4 × 4 inch pin. (25)

Here we have a variation in diameter in practice, demanded by the clearances at the front limiting the length of the pin which is accordingly made 5 inches diameter and 3¾ inches long.

As for the forms of the crank pins used on such a locomotive as this consolidation engine, they are simple and are shown in Fig. 20.

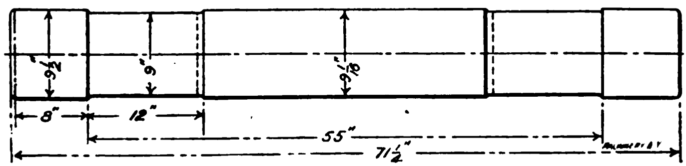

Fig. 22. Driving Axle for Consolidation Locomotive.

In this the three forms shown are for the main, intermediate and front and back pins respectively. Fig. 21 shows the method of attaching collars at the ends so as to occupy a minimum of space. As all of the rods are practically fitted with solid bearings, it is necessary that the pins should have washers at the outer ends to hold them in place.

Fig. 22 illustrates the common form of driving axles used at present. There is an enlarged wheel seat at the end. The journals are inside this and are slightly smaller in diameter than the wheel seat, while the body of the axle at the center is slightly larger. Abrupt changes of diameter are avoided and square shoulders are not allowed. The same process in the determination of dimensions may be followed in the case of the passenger locomotive.

CHAPTER VI.

DRIVING WHEELS AND COUNTERBALANCING.

The driving wheels, on which the engine is carried and by which it is propelled, are subjected to stresses far beyond those due to the mere requirements of carrying the load put upon them or transmitting the thrust exerted by the steam into rotation and pull upon the rails. Among the first in importance of these secondary stresses are those due to the shrinkage of the tires upon the center. In order that the wheels may withstand these stresses, the rim should be heavy between the spokes, and the latter stiff enough to carry the load without bending. In the shrinking on of the tire under ordinary conditions an allowance of 1/100 inch to the foot is made. That is to say, the tire is bored out 1/100 inch smaller than the diameter of the center upon which it is to be placed for each foot of diameter of the latter. This allowance is modified somewhat according to the proportions of carbon entering into the composition of both the tire and the center.

When the center is made of cast steel in accordance with prevailing practice, it should be well annealed so that the greatest possible degree of toughness may be obtained. In effecting this annealing the utmost care should be taken to do the work uniformly throughout the whole extent of the material, and the same degree of hardness obtained as far as it is possible. It is essential that this should be done in order that the application of the tire may be successfully made, for the allowance for shrinkage must be met by the compression of the center and the elongation of the tire in such proportions that the latter will be securely held in place without overstraining either.

When soft material is used either in the tire or the center, a greater allowance must be made for shrinkage than where one or both are hard. These terms are, however, indefinite as to their limits, and there is, as yet, no standard of reference by which the work has been scientifically laid down. The result is that while the figures given for the proper shrinkage are those recommended and extensively used, experience and good judgment must enter into the work, and no hard and fast rule can be laid down that will meet all of the exigencies of varying conditions. In the design of the center, the spokes should be placed as close together as practicable and should be of such a section that they can withstand, without bending, the stress required to elongate the tire, in accordance with the amount of shrinkage allowed; this stress to be divided among all of the spokes.

If we place the shrinkage at 1/100 inch per foot of the diameter of the wheel, and assume that the center compresses as much as the tire elongates, there will be a total elongation of the tire of about 1/64 inch for each foot of circumferential length, and a corresponding com-

pression of the rim of the center. It is evident, then, that in order that such a compression may take place in the rim, the spokes must be able to be compressed a corresponding amount without changing form. This requirement does not determine the form of the section of the spoke but does influence its thickness. The width of the spokes laterally should be as great as other conditions will permit, as there is no means of determining the stress in this direction that may be brought to bear upon the wheel by a rough track, a derailment or the movement at high speeds around curves and over frogs and switches. Furthermore, the wheel should be dished as little as possible, because this, with the stresses induced by the shrinkage of the tire, tends to increase the transverse stress on the spokes.

If the strength alone were to be considered, the best form of spoke section would be that of an I-beam; but as this is impracticable because cracking would occur in the cooling of the casting, the next best, or rectangular form, is used, rounded at the corners to an elliptic section. Again, an allowance must be made in the spoke lengths for rough usage and indeterminate stresses at the same time using the utmost discretion in the work lest the weights run up to excessive amounts, as may very easily occur.

The hub should be of ample strength to resist the stresses to which it is subjected by the pressing in of the axle and crank-pin. It is essential, not only that there should be sufficient thickness to resist these unknown stresses but that this thickness should be as uniform as possible so as to avoid shrinkage cracks and unequal stresses due to the cooling of the metal after casting.

In boring for the axle and pin the work should be done with the utmost care and the dimensions so proportioned that the pressure required to force the wheel and axle together will be from 12 to 14 tons per inch of diameter of the axle. Where steel or wrought iron wheel centers are used the fit should have a slight taper which will result in a better fit and less crushing of the surfaces of the material than when the fit is straight, when this may be so great as to exceed the elastic limit of the material. As a general rule the taper should be not more than 3/64 inch or less than 1/64 inch to the foot. Of the two the larger taper is to be preferred where comparatively soft steel is used in the wheel centers; while, when the steel is hard, the smaller taper should be used. The taper for the crank-pin fit may be made 1/32 inch to the foot and the pressure regulated to the same amount per inch of diameter as in the case of the axle.

There is another detail in the designing of the wheels that is of great importance and which has received the closest examination for many years, and that is the counterbalancing. To do this all parts connected to each crank-pin should be carefully calculated, or better still, weighed, if it is possible to do it before the counterbalance weights are determined.

In doing this, the side rods, the rear end of the main rod, together with all pins and straps, are to be considered as revolving parts and

the weight of each assigned to the wheel and pin with which they are connected. An approved method of weighing is to couple the side rods together and, supporting each bearing on a knife-edge with the rod horizontal, take the weights on these knife-edges in succession. The weight of the end of the main-rod is obtained in the same way. In addition to these revolving weights those of the reciprocating parts, including the front end of the main-rod, the crosshead, piston and piston-rod must be taken.

Of these the counterbalance in the wheel must be the equivalent of all of the revolving parts and a portion of the reciprocating. Regarding the latter the rules and practice have not yet taken such shape that a fixed proportion of the reciprocating parts to be counterbalanced is established. In this the designer is between two evils. The greater the proportion of reciprocating parts that are counterbalanced the smaller will be the longitudinal motion of the engine, but the greater will be the vertical disturbance. This puts an excessive pressure on the rail when the counterweight passes the lower center and tends to lift the wheel when it passes the upper; while, on the other hand, if too little of the reciprocating parts are counterbalanced, there will be an excessive longitudinal motion or nosing of the machine. As a matter of fact, an attempt is made to strike a happy medium between these extremes by which the minimum vertical effect on the rail is produced with the maximum longitudinal disturbance that can be tolerated.

To accomplish this, it is well to take the weight of the whole engine into consideration when calculating that of the counterbalance; for the relation between the two has a marked influence on the perceptible horizontal disturbance. Hence it comes about that the question has to be reversed and decided along the lines of what proportion of the reciprocating weights, that of the engine will permit to be left unbalanced.

The rule adopted by the American Railway Master Mechanics' Association is to allow 1/400 part of the weight of the engine to remain unbalanced in the reciprocating parts. This, while entirely empirical. works well in practice and is quite generally used throughout the United States. It is, however, modified in some instances by limiting the counterweights to from 55 per cent as a minimum to 65 per cent as a maximum of the weights of the reciprocating parts for road engines, though the practice in this respect is not universal. This weight is equally divided among all of the driving wheels of the locomotive. In case the wheel centers are so small that there is not room to put the whole of the counterbalance that should be apportioned to the main driver, on the side opposite the crank-pin, and get it within a reasonable area that does not exceed the area of half of the half circle, the remaining wheels should not be balanced to compensate therefor, unless the extra weight to be balanced is less than 65 per cent of the weight of the reciprocating parts divided by the numbers of wheels.

This may be expressed by the formula

$$B = \frac{0.65\,W}{n} \tag{33}$$

in which

B = the maximum allowance balance weight which is added above that needed to counterbalance the revolving parts,

W = the total weight of the reciprocating parts,

n = the number of coupled wheels on each side of the engine.

If this does not make up for the deficiency of balance of the main wheel, the total balance should be left that much short, and the maximum speed limit of the engine should be correspondingly reduced.

In designing the wheel, it is best to cast the counterweight in solid with the center. Should the weight be too great for this, pockets may be cast in symmetrically on the side opposite the crank to be filled with lead in the final adjustment, since a greater weight can be obtained in a smaller compass by the use of lead than with iron. In other cases the major portion of the weight may be cast in, and the deficiency made up by the addition of lead in the final adjustment.

This is usually done after the axle and crank-pin have been pressed in. The journals of the axle are placed upon horizontal straightedges, and the wheel turned so that the crank-pin is on a horizontal line with the center of the axle. A weight is then hung upon the pin by a ring somewhat larger in diameter than the pin so as to secure a central bearing. This weight should be equal to the sum of the weights of the ends of the side rods that will be attached to the pin, and the proportion of the reciprocating weights that is to be apportioned to the pin. In the case of the main pin the weight of the rear end of the connecting-rod is to be added.

The reciprocating weight to be balanced on each wheel may be found by the formula:

$$r = \frac{R - \dfrac{W}{400}}{n} \tag{34}$$

in which

r = the reciprocating weight to be balanced at each wheel,

R = the total weight of the reciprocating parts,

W = the total weight of the engine in pounds,

n = the number of coupled wheels on each side.

Still it must be borne in mind that this weight is subject to the conditions imposed previously, that the total weight counterbalanced or $R - \dfrac{W}{400}$ should lie between 0.55 R and 0.65 R and r will be the same for each wheel.

As for the driving boxes, their dimensions, like so many other parts of the locomotive, are not determined by any special rule except that in a general way they are proportioned according to the diameter of the journals, compounded with those indispensable requisites for good designing in all branches of mechanics, experience and good judgment.

The prime requisites are that they shall have sufficient strength to sustain the load that they have to carry and sustain the pressures to which they may be subjected in the working of the engine. Nearly all of these forces partake of the nature of compression stresses, which, of themselves, would not seriously strain the material, but they are coupled with shocks and blows both lateral and longitudinal, especially when the wedges become worn or loose in service.

Here again the importance of reducing all weights to a minimum manifests itself and so cast steel has come to be the usual material used for driving boxes. The crown brass is usually made with a thickness at the top equal to one-quarter the diameter of the axle and is forced into place by an hydraulic pressure of from 20 to 25 tons. The length of the bearing is governed by the diameter of the journal and the weight that it has to carry, and is so proportioned that on fast passenger engines the load carried does not exceed 170 pounds per

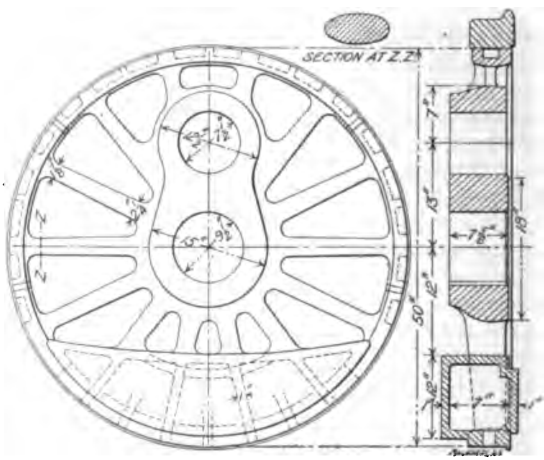

Fig. 23. Cast Steel Driving Wheel Center for Consolidation Locomotive.

square inch of projected area, and 180 pounds for freight engines, while 190 pounds per square inch can be allowed on switching engines.

In order to meet these conditions it frequently becomes advisable to increase the diameter of the axle by a fraction of an inch, above the calculated requirements, so that the bearing may be shortened and the center of the box be brought centrally beneath the springs. There should be sufficient side play between the flanges of the boxes and the wedges to permit one box to rise until it strikes the frame while its mate at the other end of the axle remains down in its normal position. The reason for this is mainly that the flanges may not be broken in case a spring breaks on one side.

The oil cellar is usually a simple cast-iron box fixed to the lower shanks of the driving-box. It is held in position by horizontal pins that may be easily removed so that the cellar may be lowered for repacking or replacing the pads, which should always be high enough so

as to press against the axle journal and thus supply lubrication for a time should the feed from the top of the box be cut off.

Turning now to the practical application of these principles, Fig. 23 illustrates a cast steel driving wheel center for the consolidation locomotive in which pockets are cast in the counterbalance for the use of lead filling, and Fig. 24 shows a wheel of the same material, but of larger diameter intended for the Atlantic engine with the counterbalance cast solid with the rim.

In the former case the weight to be counterbalanced on the main

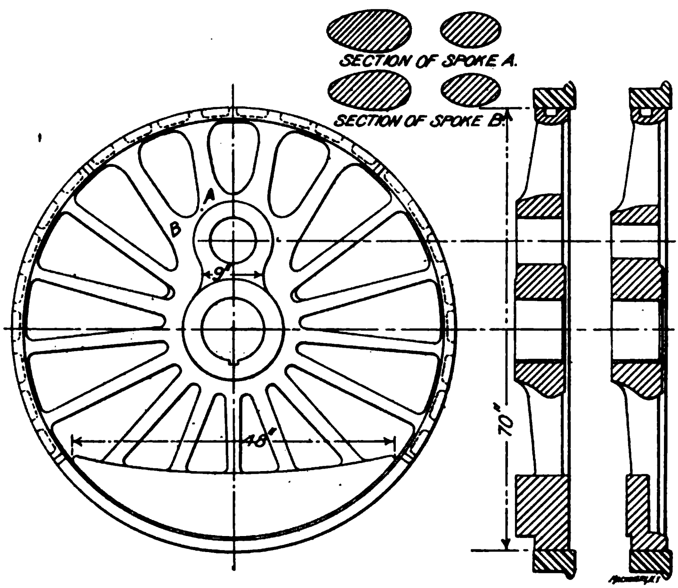

Fig. 24. Cast Steel Driving Wheel Center for Atlantic Type Locomotive.

driver is, according to the rule of the Master Mechanics' Association:

$$966 - \frac{176{,}000}{400} = 526 \text{ pounds,}$$

and for the Atlantic engine

$$1057 - \frac{168{,}000}{400} = 637 \text{ pounds,}$$

in both of which cases the weights are from 55 to 65 per cent of the weights of the reciprocating parts which are 966 pounds and 1,057 pounds respectively.

Again, if it should have so happened that there had not been room on the main wheel for this amount of counterbalance, the distribution of the weights among the other wheels would have been, according to formula (34), for the consolidation engine

$$r = \frac{966 - \dfrac{176{,}000}{400}}{4} = 131.5 \text{ pounds,}$$

and for the Atlantic engine

$$r = \frac{1057 - \dfrac{168{,}000}{400}}{2} = 318.5 \text{ pounds.}$$

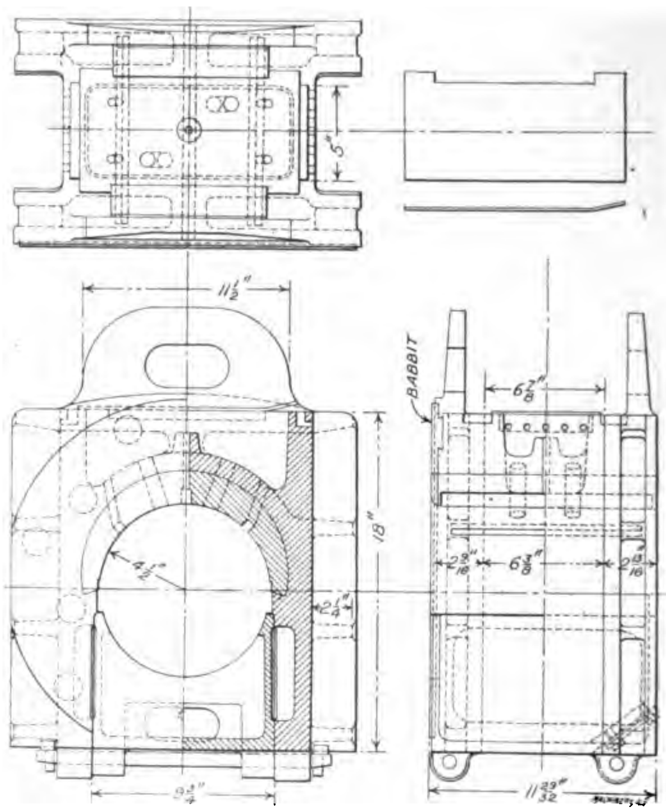

Fig. 25. Driving Box for Consolidation Locomotive.

To these must be added the weights of the revolving parts, which, aside from the crank hubs, are on the consolidation engine for

the forward pin.................................. 100 pounds
the main pin..................................... 545 pounds
the third pin.................................... 215 pounds
the rear pin..................................... 100 pounds

in the case of the Atlantic engine for

the forward pin.................................. 140 pounds
the main pin..................................... 490 pounds

All of these weights must be reduced in proportion to their distance from the center of the axle as compared with that of the crank-pin. For example, if the center of gravity of the counterbalance of the main driver of the Atlantic engine is two-and-one-half times the distance of the crank-pin from the center of the axle, the actual weight to be used becomes

$$\frac{490 + 637}{2.5} = 450.8 \text{ pounds.}$$

Fig. 25 shows the form of box that is used for the consolidation locomotive and the same general features obtain in that for the Atlantic. Particular attention is called to the form of the flanges, where they bear against the wedges. They are tapered from the center to the ends, the distance between them widening, so that they can have a rocking motion without binding, and thus fulfill the conditions imposed that the box shall be able to rise and strike the frames and not bind while its mate is in the normal position. It will be seen that the flanges are quite heavy and that the depth is sufficient to give a good bearing on the wedge faces. The length of the bearing is made 11¾ inches long, and as the diameter of the axles is 9 inches, this makes 105¾ square inches of projected area of bearing. As the weight on the drivers is 19,375 pounds, from which must be deducted the weight of the wheel and one-half the axle, which will amount to at least 3,000 pounds, the load on the journal falls within the limit of 180 pounds per square inch, or will be about 155 pounds per square inch, a margin that leans to the side of safety and cool running.

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The foregoing books, up to and including No. 26, were published and in stock in November, 1908. The remainder will go to press as rapidly as practicable. The complete plan of the series, as stated, is to cover the whole field of mechanical practice, and the editors are preparing the additional titles, which will, from time to time, be announced in Machinery.

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MACHINERY'S REFERENCE SERIES

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No. 30

LOCOMOTIVE DESIGN

By Geo. L. Fowler and Carl J. Mellin

PART IV.

SPRINGS, TRUCKS, CAB, AND TENDER

CONTENTS

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CHAPTER I.

SPRING RIGGING AND EQUALIZERS.*

The proper distribution of the weights on the wheels should be considered as soon as the type of engine and the service which it is to perform have been decided upon. The weight that can be placed upon each wheel is limited by the rail, the bridges on the line and the general features of the track construction. In this distribution of the weight the method of spring suspension and equalization plays an important role, and it must be so designed that a safe load and one that will hold the truck wheels down on the rails under all conditions of speed and track curvature is put on them, while that on the driving wheels must be sufficient to give them the proper adhesion for the development of the required power.

It may be stated that in a general way the number and size of the driving wheels are chosen in connection with the allowable weights on the same according to the service performed, while the size of the cylinders is determined by the weights and dimensions chosen. The cylinder dimensions and boiler pressures are so proportioned that the power exerted at the rail, which is expressed by the following formula, shall fall between 22 and 23 per cent of the total load on the driving wheels:

$$T = \frac{d^2 \times 0.85\, p\, S}{D} \qquad (1)$$

in which

T = the tractive power,
d = diameter of cylinders,
p = the boiler pressure,
S = stroke of piston in inches,
D = diameter of driving wheels in inches.

When these figures have been decided upon, a skeleton drawing of the engine is prepared by which the weight and its distribution is outlined, and here the locomotive designer's judgment and experience is called into play. The layout is afterward carefully checked by a calculation of the weights of the various parts and their moments with reference to the center of the truck when a four-wheeled bogie is used; otherwise they are taken from the center of the cylinder.

* The present number of MACHINERY'S Reference Series is the fourth part of a treatise on complete Locomotive Design, covered by Nos. 27, 28, 29 and 30 of the Series, and originally published in RAILWAY MACHINERY (the railway edition of MACHINERY). Each of the four parts of the complete work treats separately on one or more special features of locomotive design; and while the four parts make one homogeneous treatise on the whole subject, each part is complete by itself. In order to give concrete form to the examples and theoretical considerations, it is assumed that a consolidation freight locomotive and an Atlantic type passenger engine are being designed. It is further assumed that these locomotives are designed for a division 150 miles long, laid with rails weighing 75 pounds per yard, and with a ruling grade of one per cent ten miles in length.

When the desired weight of engine has been obtained, the wheels should be so arranged that they form two groups or systems of supports. The one at the front should be cross equalized so that the center of support falls in the center line of the engine. This is very readily accomplished where a four-wheeled truck is used at the front and takes care of the front system. All of the remaining wheels are then included in the system for the rear. When these can be equalized on each side so as to form three points of suspension, ideal conditions have been obtained. This can be readily accomplished in the case of eight or ten-wheeled engines.

When the rear system is entirely composed of driving wheels, as in the case of the engines just mentioned, the momentum of the load falls along the line of the center of gravity of the axle support. But, when trailing wheels, carrying a lighter burden, as in the case of the Atlantic engine, are used, it falls along a resultant line of these several centers. It is then necessary to so adjust the wheel base that the weight is carried on a line coinciding with the resultant axle centers of gravity.

In the case of mogul or consolidation locomotives where a two-wheeled truck is used at the front, one or two pairs of the forward drivers are equalized with it. The common supporting point of the system is brought to the center of the engine by a transverse equalizer, usually placed on a line with the front end of the forward driver springs, and this coincides in its longitudinal center, as it crosses the engine, with the center of gravity of that portion of the machine and load that it is proposed to carry on the forward system of suspension. The center of the load of the rear system falls, as in the previous case, in the center of its wheel base if the wheel spreads are equal; otherwise in the center of gravity of its respective axle supports.

With these points known, as well as the distance between them, together with the weight of the engine above the axles, the design should be modified by the shifting of the boiler and other parts, so as to bring the load on the proper centers in a way to secure the desired distribution on the axles. The weights of the several parts should be calculated with all possible accuracy, especially where this data regarding the parts of an earlier engine are not available for comparison; and for this work no short-cut rule can be given.

The weight on the axles having been determined, the next point is to ascertain the size of the springs to carry the load, which is equal to that on the axles, less the weight of the driving boxes, spring saddles, and the springs themselves. This is found by means of the following formula:

$$P = \frac{S\,b\,h^2 n}{6\,l} \qquad (2)$$

in which

P = load on one end of the spring,

S = allowable fiber stress in the steel, usually put at 80,000 pounds,

b = width of spring leaves,

h = thickness of spring leaves,

n = number of spring leaves,

l = length of spring from edge of spring band to point of spring hanger bearing. as indicated by Fig. 1.

In this P is, of course, equal to one-half the total load on the spring.

For ease of riding, the leaves should be made as broad as possible, and experience shows that a length of from 36 inches to 42 inches from one point of suspension to the other is that best adapted to locomotive work, though this is, of course, to be governed by circumstances and the conditions surrounding the design, which may involve the use of either a longer or shorter spring. As for the thickness of the leaves, that varies from 3/8-inch to 7/16-inch.

The spring when first made must have a certain amount of free height or set, so that, when the load has been applied, it will deflect to a point best adapted to the carrying of the load. This deflection may be found by the formula:

$$F = \frac{2 P l^3}{E b h^3 n} \qquad (3)$$

in which

F = the deflection,

E = modulus of elasticity of the steel, which may be put at 31,500,000.

The other symbols have the same significance as in the case of formula (2).

For the helical springs that are frequently used at the extremities of the spring suspension, the total carrying capacity is calculated by the formula:

$$P = \frac{S \pi d^3}{8 D} \qquad (4)$$

in which

d = diameter of steel of which the spring is made,

D = diameter of coil to center of steel.

The other symbols are the same as in formula (2).

The diameter of the steel to be used thus becomes:

$$d = \sqrt[3]{\frac{8 P D}{\pi S}}$$

The deflection of the helical springs will be

$$F = \frac{8 P D^3}{\pi G d^4} \qquad (5)$$

in which G is the modulus of elasticity for torsion, and may be put at 12,000,000, or

$$F = \frac{D S l}{d G} \qquad (6)$$

Finally the length of the wire between end coils will be found by the formula:

$$L = \frac{F G \pi d^4}{8 P D^3}, \text{ or } \frac{F d G}{D S} \qquad (7)$$

in which the significance of the symbols remains as before.

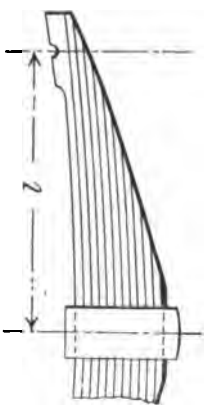

Fig. 1. Diagram for Use with Formula (2).

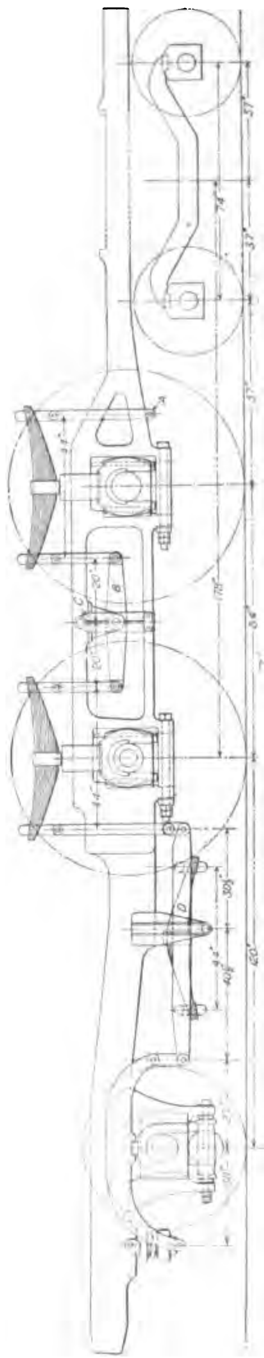

Fig. 2. Spring Rigging for Atlantic Type Passenger Locomotive.

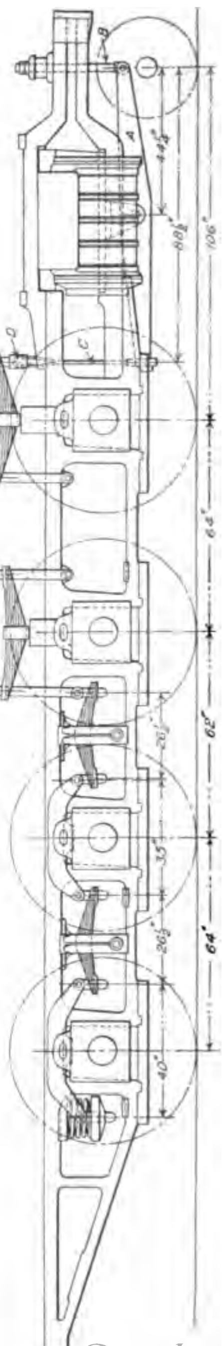

Fig. 3. Spring Rigging for Consolidation Type Freight Locomotive.

Figs. 2 and 3 illustrate the manner in which the principles that have thus been laid down have been followed in the case of the two engines, the designing cf which we are here following. On the Atlantic locomotive (Fig. 2) there are two systems of equalization: One at the front cared for by the four-wheeled truck, and which is not shown in detail, as the load is carried on the center plate and needs no further explanation. In the rear system, there are four points of support for the frame. Starting at the front, the forward spring hanger carries the frame at the point *A* and transmits the load sustained at that point to the semi-elliptic spring set on the axle box. A hanger dropping down from the rear of this spring takes hold of the equalizer *B*, which is pivoted at its central point on the fulcrum *C* and serves to sustain its portion of the load.

In the same way the stresses are transmitted to and through the spring over the rear driver and down to the equalizer *D*, which forms the connection between the rear driver and the trailing truck wheel. At this point the overhang of the firebox makes it impossible to place the semi-elliptic springs above the frame, so the equalizer is made of two bars, and a spring shorter than the distance between the supporting hangers is used. This is also done in order that the spring may not be of excessive weight and length. The rear axle box is fitted with a yoke, at the back end of which the helical spring for supporting the rear frame is placed. In this suspension every point of support is a pivot, and there are no rigid connections. Hence there can be no variation from the proportioned distribution of the weights.

For example, it is evident that the load carried by the forward spring hanger at *A* must be equal to that on the rear hanger of the same spring otherwise the spring would be tipped down toward the hanger with the greatest load upon it. This would extend back through the whole system of suspension. If such a tilting should occur toward the front, for example, that action would then tilt the intermediate equalizer *B*, which would put an additional stress upon the forward hanger of the rear driver spring. Such an additional load would be transmitted on to the helical spring at the back, which would at once resist further compression by adding to the weight it was normally lifting and thus check any tendency of the whole system to tilt by sending forward to the point *A* a greater lifting power and thus, by increasing the stresses on the forward hanger, equalize the system and distribute the proper proportion of load among all of the points of support.

In order to perform the supposititious work of this engine, as stated in the foot-note on page 3, one weighing 167,000 pounds, with 23,000 pounds on each driving wheel, would have to be offered. This leaves 75,000 pounds to be distributed between the trailing wheel and the front truck. In proportioning the rear equalizer, the two arms have been so designed that a load of 18,500 pounds is put upon each of the trailing wheels, leaving 38,000 pounds to be carried by the front truck. These weights include those of the wheels, so that the actual load on the springs will be less by the amount of the weights of the wheels, axles,

boxes and springs. As the total of these weights will amount to about 13,000 pounds for the driving wheels, with their axles and boxes, and 9,000 pounds for the trailing wheels, with the same connections, the springs with the arrangement shown must provide for sustaining a total load of 107,000 pounds. As part of the load is carried twice, so to speak, due to the equalizing arrangement, the various springs will be loaded as follows:

Driver box springs, each	19,750 pounds
Intermediate driver and trailing truck equalizer spring	17,275 pounds
Trailing truck helical spring	6,600 pounds

In these weights the total load on both ends of the spring is given. so that in the calculation of the same, the figures must be divided by two as already stated.

The suspension for the consolidation locomotive is shown in Fig. 3. Starting with the forward driver, equalization is accomplished by means of the equalizer *A* with a bearing on the bottom of the saddle casting on the center line of the engine. At the front this equalizer is suspended and pulls down upon a bolt *B* whose upper end rests upon a spring cap that is carried by the truck. At the rear it pulls down on the hanger *C*, that is attached to the center of the cross-equalizer *D*. whose outer ends are carried by the hangers coming down from the front ends of the springs. The balance of the equalization of this system is clearly shown by the engraving.

As for the rear system, which includes the three pairs of drivers, it resembles that of the Atlantic engine, except that the springs are of a sufficient length to reach from one driver box yoke to the other, thus taking the place of an intermediate equalizer. In this engine the weight on the drivers may be given as 19,500 pounds each. With the proportions of the forward equalizer shown in Fig. 3, the weight on the two truck wheels would be about 21,000 pounds, so that the total weight of the engine would be 176,000 pounds. By making the same reduction as before for the wheels, axles, and boxes, the weight to be carried by each of the springs becomes 16,300 pounds and they may be calculated accordingly.

The spring strengths are not the only calculations that have to be made in the work of the suspension. This also involves that of the equalizers and hangers. The former can be calculated from the formula for beams supported at the ends and loaded in the middle as follows:

$$b\,d^2 = \frac{18\,P\,L}{S} \qquad (8)$$

in which

b = the thickness of the bar,

d = the depth of the bar,

L = distance between hanger connections in feet,

S = safe fiber stress = 8,000 pounds for steel castings. and 14,000 pounds for wrought ircn,

P = load to be carried.

Fig 4 gives the general proportions for these parts for the Atlantic engine. It is necessary, however, in the designing of these parts to make them of ample strength to withstand the shocks to which they will be subjected, hence the low fiber stress that has been specified in the schedule for formula (8).

The spring hangers must also be designed of ample strength; these are not only subjected to a tensile stress that may be applied with more or less suddenness but one which is also constantly varying when the engine is in motion. In this, too, the fiber stress should not be allowed to exceed 8,000 pounds for the static load. Exactly what it will

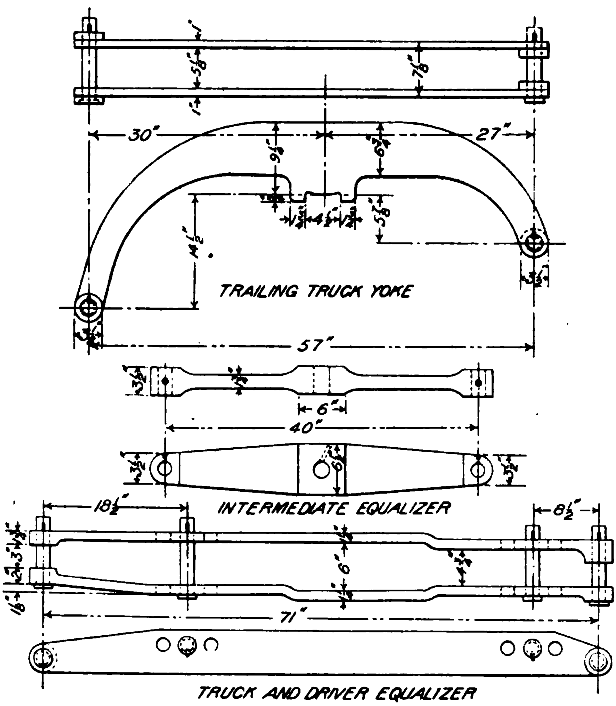

Fig. 4. Yoke for Truck Box and Equalizers, Atlantic Type Locomotive.

amount to when the engine is in motion is not known, but it is apt to be as much as 50 per cent in excess of the calculated load. Fig. 5 shows a number of types of spring hangers that are used on consolidation and mogul locomotives, including not only those that are to be found upon the engine under consideration, but some others as well.

In the whole of the spring suspension, as in other parts of American cars and locomotives, nothing is fastened down, but dependence is placed upon the weight to hold the parts in position. The hangers, therefore, rest upon the ends of the springs through the intervention of the hanger gibs, some of the forms of which are shown in Fig. 6. For these no

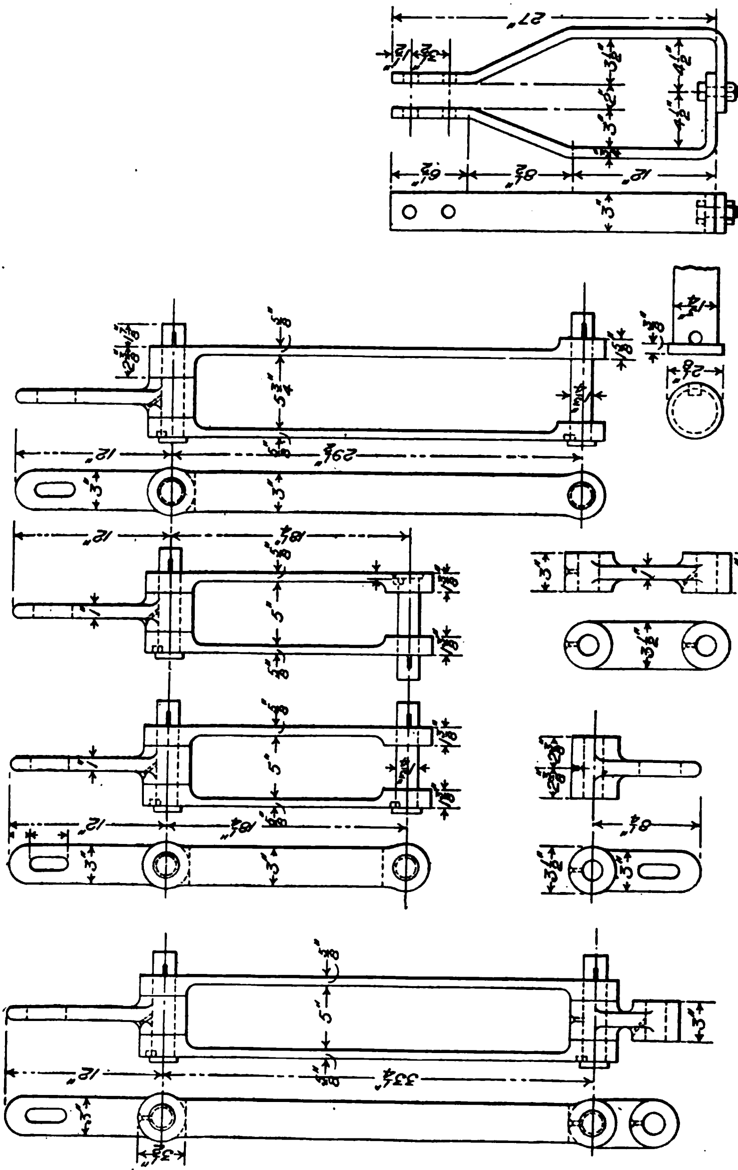

Fig. 5. Spring Hangers for Consolidation Locomotives.

calculation is to be made. They are subjected to a shearing stress only and must be made heavy enough to withstand this and allow for a large amount of wear as well. As the latter is the larger item of the two, the actual stress that is put upon the gibs when the engine is new and everything up to its original dimensions, is very small.

It will thus be seen that for a proper adjustment of the spring suspension, great care must be taken so that the center of gravity of the portion that is to be carried by the front or back system of suspension should fall in the proper place. In the case of the Atlantic engine, we have the two driving wheels located 84 inches apart and each carrying 23,000 pounds. The trailing wheel is 120 inches behind them and carries

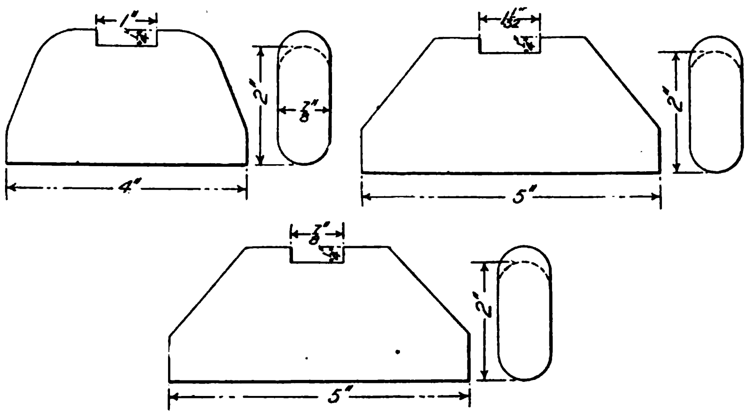

Fig. 6. Spring Hanger Gibs.

18,500 pounds. The center of gravity of these three weights falls 0.4 inch back of the rear driver, and that is the point at which the center of gravity of the weight to be carried should be arranged to fall. The same care should be taken in the case cf the consolidation locomotive that the center of gravity of the weight carried by the rear system should fall in a line with the next to the rear pair of drivers when these are spaced equal distances apart.

When these precautions are taken, the engine will nct only be easy riding, but there will be far less trouble with the parts of the spring suspension in the way of breakages than where these precautions are ignored.

CHAPTER II.

THE TRUCKS.

The truck is an essentially American characteristic of the locomotive. For many years European locomotives, especially those used in freight service, were built without any truck, and the guiding of the engine was done by the flanges of the front pair of wheels. In this country, however, the truck has always been used upon road engines and has been considered an essential detail in their safe and satisfactory operation.

Broadly speaking, the engine truck proper, or the one located at the front end, may be divided into two classes, the two- and four-wheeled types. The four-wheeled truck is the one that has been universally used on locomotives intended for passenger service, while the two-wheeled has been applied to freight engines, or those that are used, for the most part, in freight service, with an occasional assignment to passenger work. The exclusion of the two-wheeled truck from engines designed for passenger service has been due to the necessity of using a large boiler and so increasing the total weight of the engine beyond the requirements of adhesion represented by the tractive power that it was desired to develop. Under these circumstances the extra weight, beyond that needed for adhesion, could be carried on four wheels to better advantage than on two. As for safety, there is no difference of opinion that the two-wheeled truck is quite up to all the demands of the most exacting service.

In the designing of the trucks there is little else to be done than to secure ample strength to carry the load imposed and arrange for axles and boxes of sufficient bearing surface to do the work required without heating. At the same time care must be taken that sufficient weight is put upon the truck to hold it down and cause it to keep the rails upon the sharpest curves to be encountered, and thus prevent the flanges of the wheels from climbing the rail and causing a derailment when called upon to guide the direction of motion of the machine from a straight line to a curve and through the latter. In the case of the consolidation engine under consideration, as well as upon those of the mogul class, the two-wheeled pony or Bissel truck is used.

The plan of framing this type of truck is shown in Fig. 7, and the details of the working parts in Fig. 8. From these drawings, as well as from a comparison with Fig. 3, it will be seen that the truck is pivoted at the center between the wheels. As this would not hold them in place on the rails, but would allow them to swing into a position approximately longitudinal to the track, if any obstruction were to check the motion of one, a radius bar, A, is added that reaches to a point beneath the engine, where it is pivoted on the center point. This

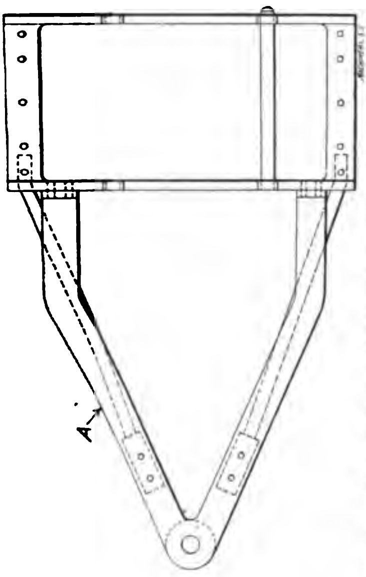

Fig. 7. Outline Plan of Pony Truck Frame for Consolidation Locomotive.

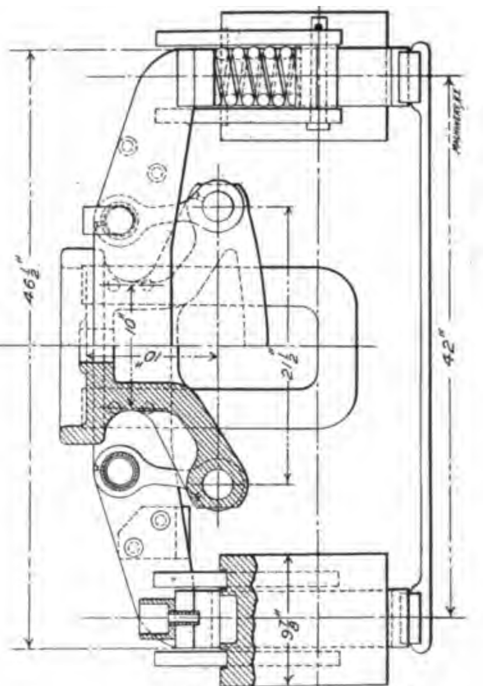
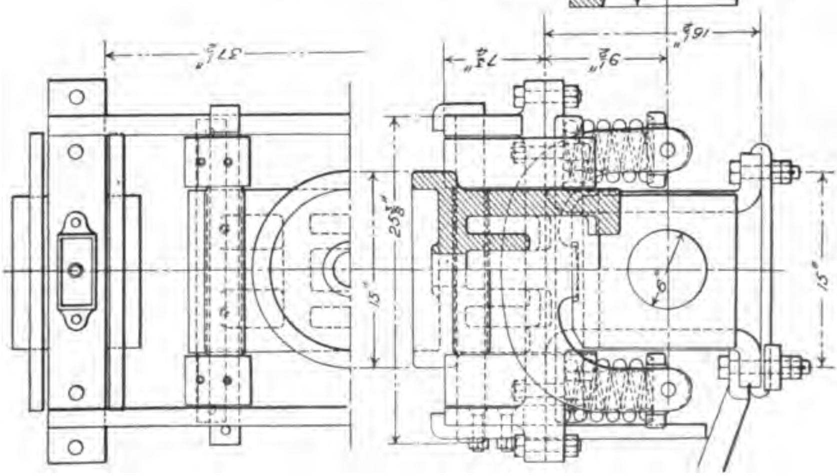

Fig. 8. Pony Truck Frame for Consolidation Locomotive.

point is carried back far enough to insure stability of the wheels upon the rails, and, at the same time, permit of sufficient side motion to allow the truck to swing out of line with the center of the engine when entering and passing over a curve.

The formula used for calculating the length of the radius bar is as follows:

$$R = \frac{A \times B}{A + B} \times 0.85 \tag{9}$$

in which

$R =$ the length of the radius bar,

$A =$ the total wheel-base for consolidation and mogul locomotives,

$B =$ the distance from the front driving wheel to the truck wheel.

In the case of the consolidation locomotive under consideration, $A = 314$ inches and $B = 124$ inches. The formula (9) therefore becomes:

$$R = \frac{38,936}{438} \times 0.85 = 75.56 \text{ inches.}$$

This, then, may be taken as the distance from the pivotal point of the radius bar to the center of the truck axle, which in this case is made 6 feet 3½ inches.

As the radius bar has no load to sustain and the only stress to which it is subjected is that of holding the wheels on the track, it is usually made of a flat bar of steel about 5 inches by 1¼ inch laid flat and stiffened by round braces rising diagonally from the foot of the pedestals and bolted to the horizontal portion of the bar itself at a convenient distance back of the truck frame.

The design of frame, as shown in Fig. 8, may be taken as typical of that in use upon mogul and consolidation locomotives in the United States, and is of an exceedingly simple construction. The weight of the front end of the engine is carried on the center plate, and this in turn is suspended by the center plate hangers from the transoms that reach across from side to side. These hangers are spread a small amount at the bottom so as to increase the tendency of the truck to return to the central position when coming back to a straight line from a curve. As the center plate and the pivot pin of the radius bar are normally located in the center line of the truck, it would be merely an extension of the rigid wheel base if no lateral flexibility were given to the wheels. It is this lateral flexibility, and the pull on the center plate by the hangers that tends to guide the front of the engine out of a straight line and around a curve. In the case of the truck under consideration, the frame is carried by two helical springs at each side, and these, in turn rest on seats attached to yokes that set on top of the axle boxes.

We have already noted that the weight on the truck of this engine is to be about 21,000 pounds, or 10,500 pounds on each wheel. In order to carry this load an axle 6 inches in diameter is provided with a journal 9⅞ inches long. This gives a load of about 177 pounds per

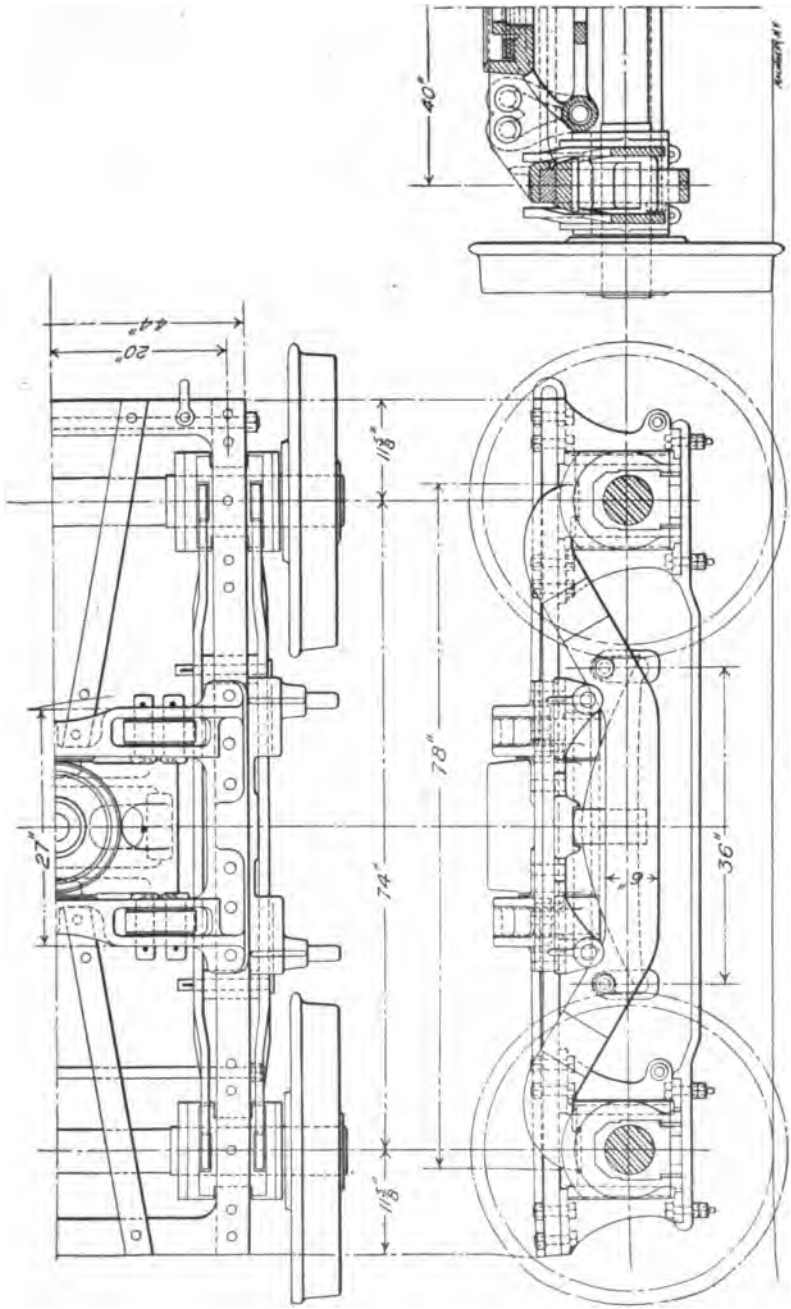

Fig. 9. Four-wheeled Forward Truck for Atlantic Locomotive.

square inch of projected area, or somewhat less than the 180 pounds usually allowed for the driving axles of freight locomotives.

The ordinary four-wheeled front truck of locomotives used on the eight-wheeled, ten-wheeled and Atlantic types calls for little or no calculation that is of value other than the determination of the strength of the equalizing bars from which the semi-elliptic springs, upon which the frame rests, are suspended. The front end of the engine rests, through a center plate on the saddle, upon the center plate of the truck. In the particular truck illustrated in Fig. 9, the center plate is suspended from the transoms by hangers in the same manner as in the case of the pony truck for the consolidation locomotive. There is a difference in the form of the hangers, however, in that, in this case, the hangers have two bearings at the top, arranged so that the two on either side act as inclined hangers, and yet remain parallel to each other; this tends to keep the center plate in a horizontal position.

In the calculations of the weight to be given to this Atlantic type locomotive, the total is 167,000 pounds, of which 92,000 pounds are upon the driving wheels. Of the balance, about 38,000 pounds will be upon the front truck and 37,000 pounds upon the rear. This puts a load of 9,500 pounds on each of the four forward wheels, with which a 5¾-inch axle is used having journals 12 inches long. This gives a pressure of somewhat less than 140 pounds per square inch of projected area, when allowance is made for the weight of the wheels, or less than the amount allowable on the driving journals of a passenger locomotive. The wheel base of these four-wheeled trucks averages from 6 feet to 6 feet 6 inches. This is a matter that is subject to change due to local conditions such as the diameter and arrangement of the cylinders, the position of the forward pair of drivers, the proportion of weight that is to be carried on the truck, and other items for which no rule can be laid down, and for which the designer must depend upon his own experience and knowledge of the fitness of things.

The rear truck of the Atlantic locomotive is of special design and varies with the builder. This type of locomotive has led to several designs of trucks for this point, most of which have been patented by the locomotive builders or individuals connected with the railroad service. It is essential that the wheels should have a lateral flexibility of movement, as in the case of the pony truck of a consolidation locomotive, in order that the virtual length of the rigid wheel base may be kept down to the actual length, or the distance between the driving wheel centers. It is also desirable that the center line of the axle shall remain as near radial to the curve of the track as possible. The trucks used in this place are designed with such an end in view. As in the case of the other trucks, no formulas can be given for the determination of the dimensions of the several parts, other than the ordinary ones in use for beams and similar structures.

In the truck, illustrated in Fig. 10, the axle is 8 inches in diameter with a journal 14 inches long. As this has to carry a load of approximately 18,500 pounds at each end, the load per square inch of projected

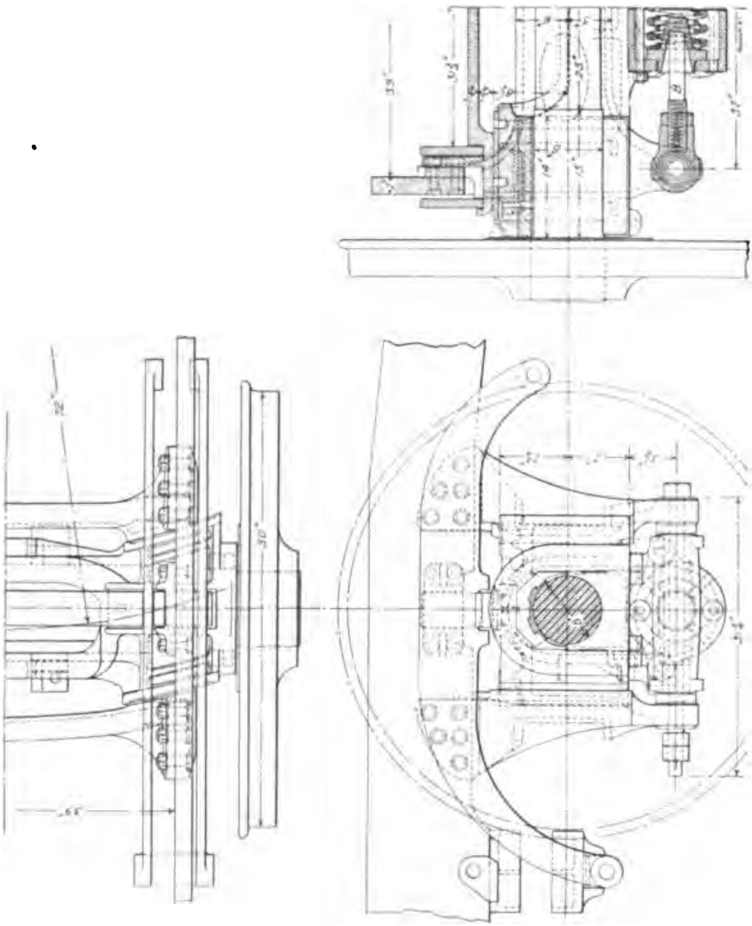

Fig. 10. Radial Trailing Truck for Atlantic Locomotive.

area is less than the 170 pounds allowable on passenger driving axles when due allowance is made for the weight of the wheels.

The axle is, of course, straight; but the boxes, instead of having their wearing surfaces parallel to its center line, are set in pedestal jaws that are cut at an angle and on a radius of 6 feet. Hence, the transverse movement of the wheels is the same as though they were swinging through the arc of a circle whose center is at the center of curvature of the faces of the pedestals. This holds the wheels at an approximation to a truly radial position, but not exactly. The swing of the overhang of the engine beyond the rigid wheel base throws these wheels out of center, and the return of the frame to its central position on a straight track would restore the normal condition of things, but assistance is rendered in overcoming the resistance of the

boxes to this movement by a centering spring placed beneath the axle. The pedestals are, of course, rigid transversely with the main frame of the engine and carry the push bars *B*, that bear against followers that are set in castings that move laterally with the boxes. Any movement of the latter, in either direction, compresses the intermediate spring, which thus has a constant tendency to restore the truck to its central position.

It will thus be seen that but very little of a definite guiding nature can be said regarding the designing of the trucks that are to be used under American locomotives. The general design of the pony and the four-wheeled truck has been established by such long usage that it is no longer a subject for discussion. Details are being constantly varied to meet the changing demands of weight, proportions and service, but with no essential change in the main features of the designs.

With the rear truck of the Atlantic, Pacific and Prairie types of locomotives, the case is somewhat different. These locomotives have not been in service long enough to have settled down to an established basis of construction in detail and it will probably be a number of years before this will be done. Meanwhile the general principle of the freedom of lateral movement for the wheels, with the approximation to the maintenance of a radial position for the axle has been accepted and acted upon, and beyond this the truck may be said to be in a process of development, though the engines to which it is applicable may be considered as among the standards of American railroad practice.

CHAPTER III.

CAB, CAB FITTINGS, AND ACCESSORIES.

As the cab contains all of the various means that are required for the operation of the engine, the greatest care is exercised to so place them that they are all of easy access to the operator. No absolute rule can be laid down for the arrangement of these several fittings, except to say that, in a general way, the planning is to be done for the convenience of operation and the comfort of the men in the cab. It, therefore, becomes largely a matter of good judgment on the part of the designer or of the choice of the purchaser, and is further dependent, to a great extent, upon the space available and other conditions appertaining to the form and construction of the appliances that are to be used. These include the injectors, lubricators, engineer's brake valve, gages, etc., which are usually bought from manufacturers who make a specialty of such supplies.

Formerly the cab was invariably built of wood, strongly braced, and so constructed that it could be lifted from its place and set aside for painting and repairs while the engine itself was in the repair shop. The advantages claimed for the wooden cab were that it was warmer in winter, cooler in summer, cheaper in first cost, and more easily repaired in case of accidental damage. Whether the claims could be substantiated or not, the fact is that now the steel cab is used generally upon new work. The cab is usually made of plates about ⅛ inch thick, and is strongly braced with angles and tees. In the case of the two engines that have been under consideration, there is but little difference in the size of their cabs. Some of these differences are due to the character of the engine and some to the preferences of the user. In the case of the consolidation locomotive, the cab has a body length of 7 feet, with a rear roof projection of 3 feet 6 inches. These are approximately the standard dimensions of the modern cab, and allow it to set over the boiler for a sufficient distance to house the cab attachments and levers, and still leave ample room at the back for the footplate accommodations for the fireman. This projection of the boiler back into the cab is usually from 48 to 50 inches at the top. In the case of the consolidation locomotive, it is 49 inches. Where the back head of the boiler is sloping, as it is in the Atlantic locomotive, it is sometimes necessary to place the bottom of the firebox farther back in the cab, in order to have room on the top for the necessary attachments. This was the case in this instance, and the cab is 6 inches longer than that of the consolidation, in consequence.

The corners, bottom edge, carlines and plates are made of angles riveted to the sheets, and it is common, though not universal, practice to countersink the heads on the outside so as to obtain a smooth sur-

face. At the front there is usually a door upon each side giving access to the running board, though sometimes this is replaced, on the right-hand side, by a drop window. The sides are invariably provided with windows running the whole length of the cab and arranged to suit the user. The sashes are made to slide past each other so that the actual point of opening can be varied. The side opening in the consolidation locomotive cab that is here illustrated is divided into two parts. The front space, which is 23½ inches wide in the clear, is filled by a fixed sash, while two sliding sashes are placed in the space at the rear. These move past each other and make it possible to have the double space clear. In the Atlantic engine, the more common arrangement of one fixed and one sliding sash is used. The fixed sash is placed at the front and the sliding sash moves across inside of it. This gives one wide opening for the engineer to lean out of.

Cab Roof.

The roof is arched from eave to eave and is carried by carlines made of angles. It is of sheet metal, as in the case of the consolidation, or of wood as in the Atlantic engine. Wood is preferable for the roof on account of its freedom from the annoyance of sweating or moisture in cold weather. The roof is exposed on the inside to the steam that occasionally rises from the boiler and its attachments, and when becoming cold, this steam is apt to be condensed on the lower surface of the roof and fall back in large drops to the annoyance of the crew.

With the boiler projecting back into the cab for a distance of 4 feet or more, it occupies so much of the cubic contents of the latter that it would heat the air to an uncomfortable extent were the cab roof not provided with a ventilator. This is, therefore, always done at present. It consists of a trap that opens upward on a hinge at its front end, or of a sliding hatch. The trap is to be preferred because its position is such that, when the engine is in motion, it not only deflects such cinders as may strike the roof from the stack, but it has a tendency to assist in drawing the hot air out of the cab. The cab is supported by heavy brackets attached to the frames, and is entirely self-contained.

Throttle and Reverse Lever.

Within the cab there are but two attachments that belong to the locomotive itself and are so necessary for its operation that they are included in the basic design for the engine. These are the reverse and throttle levers. The reverse lever is usually pivoted in a bracket bolted to the frame, and works in or beside a quadrant notched to receive its latch. As the point of cut-off is varied by the movement of the reverse lever to and fro, it is desirable that the notches in the quadrant shall be as close together as possible so as to permit of small variations in the point of cut-off. For this reason they are frequently cut to resemble a straight-tooth gear and the lever latch is made with a number of teeth to mesh with them.

It is essential in designing the reverse lever that it should be made of a strength sufficient to resist the jars and jerks to which it will be

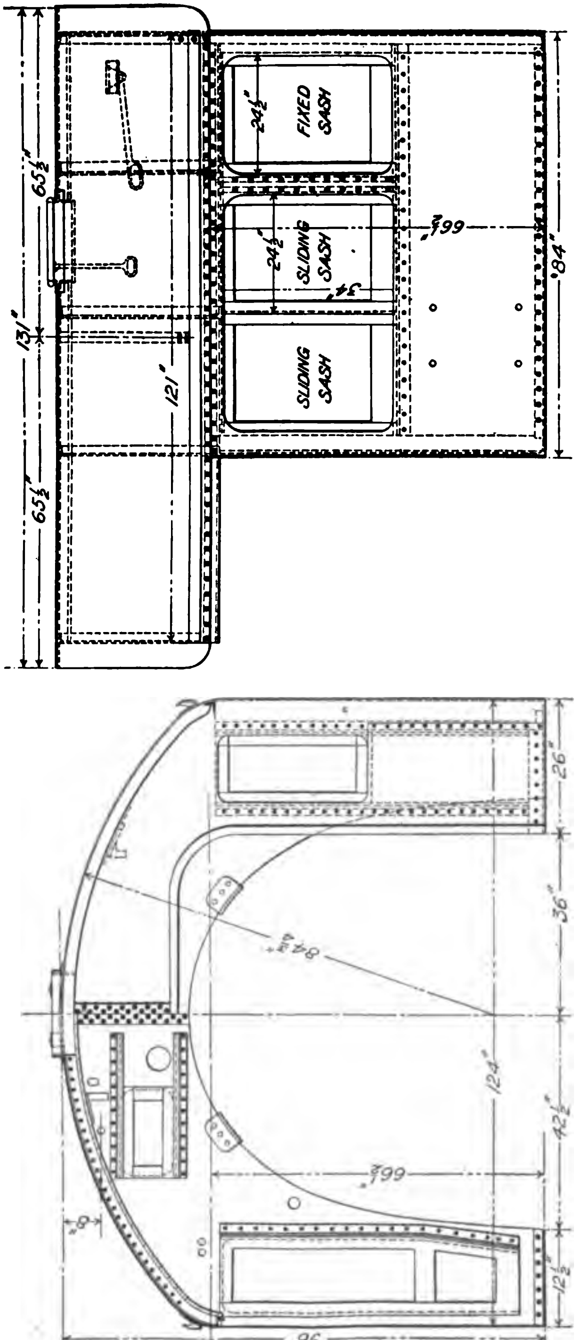

Fig. 11. Steel Cab for Consolidation Locomotive.

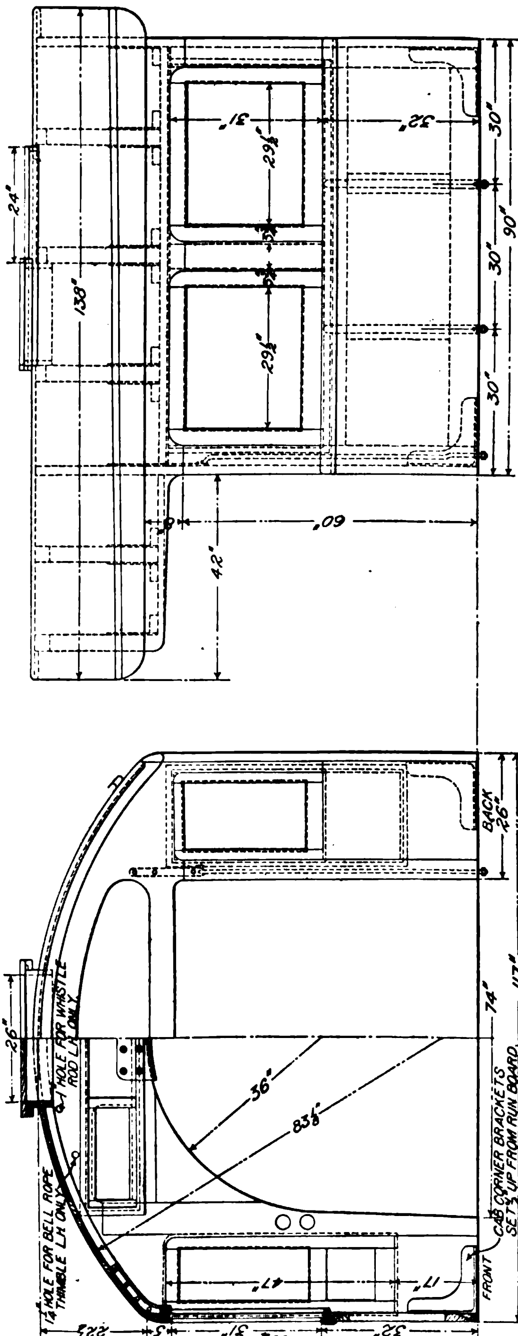

Fig. 12. Steel Cab for Atlantic Type Locomotive.

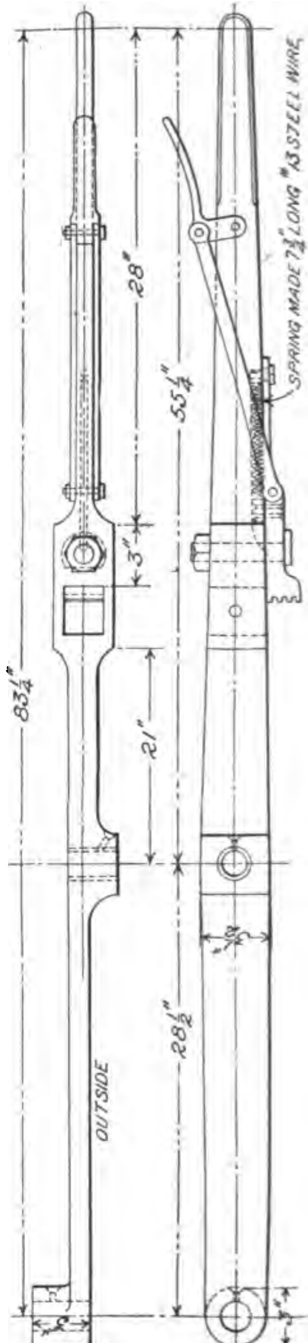

Fig. 13. Design of the Reverse Lever.

subjected, and also have a length sufficient to enable the engineer to move it, and thus shift the point of cut-off when the engine is in motion. The eccentric straps of a locomotive have been likened to a pair of prony brakes having a constant tendency to throw the links down into the position of full forward gear. So, as there is nothing to resist this tendency when the latch is lifted from the quadrant, but the pull of the man on the lever, the leverage should be sufficient to enable him to do it without danger of having it jerked from his grasp, or of being pulled forward himself. A leverage of about three to one, as in the lever illustrated, will usually be found to be sufficient for this purpose.

Especial attention must also be paid to the latch. It must be cut square and fit the quadrant with little or no sloping sides, to prevent it jarring out. The spring that holds it down in place must be stiff, and there must be such ample leverage on the latch lifter that it will be easily operated. When the engine is working, there is a constant and severe jarring of the lever as the pull of the links tends to throw it forward, and unless it is rigid in itself, and the latch is held down

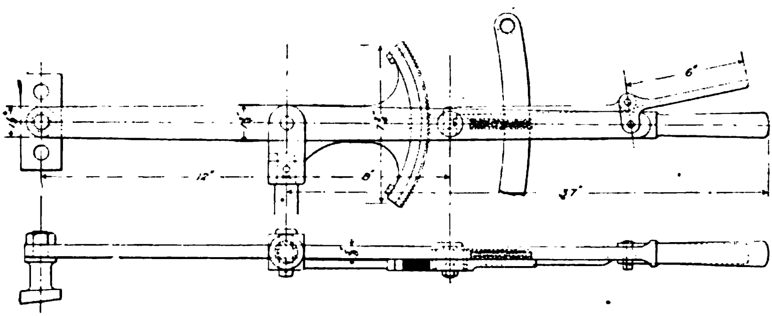

Fig. 14. Throttle Lever for Consolidation Locomotive.

firmly by the spring, it is likely to work itself free, and in its movement forward may inflict severe injury upon the engineman or the machinery.

The throttle lever should be carefully designed so that it will hold the valve in any desired position. Strength is required in order that it may be able to hold the valve firmly down upon its seat, and not because it is otherwise subjected to any severe stresses. When the valve is open, the load upon the lever sinks to insignificance; and, as the valve is usually balanced, the leverage required for manipulation is not great. It should, however, be provided with a latch that admits of fine adjustment and will hold it secure when the valve is shut. The quadrant had best be made of wrought iron, case-hardened and cut with teeth of fine pitch. Good results can be obtained if there are about six to the inch. It will be noted that, in the design illustrated, which is the one in common use, the quadrant moves with the throttle stem, so that the effect is to make the possible adjustment even less than that corresponding to the pitch of the notches. The length of leverage indicated not only gives the engineer a very delicate and

sensitive control of the valve, but is, in part, necessitated by the fact that the throttle stem is in line with a vertical plane passing through the center of the boiler and it is necessary to bring the handle out on one side to be within easy reach.

Running Boards.

The running-boards are closely allied to the cab, and frequently extend back into it, forming the foot-boards. They are of simple design; as shown in the cut, Fig. 16, they are made of steel plate about 3/16 inch thick, and are stiffened at the edges, usually by a 2 × 2-inch angle. They must be cut away where necessary to allow for the placing of pipes, air-pump, and other fixtures. As tne load on them rarely

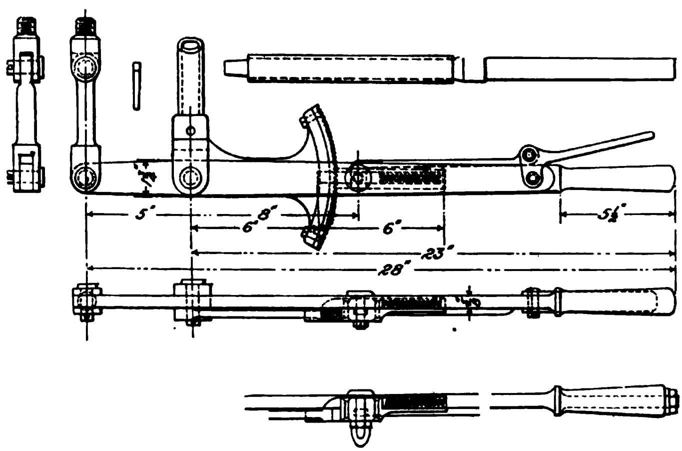

Fig. 15. Throttle Lever for Atlantic Type of Locomotive.

exceeds the weight of a few men, they are readily carried by simple brackets of flat iron riveted to the boiler. They should reach from the cab to the steam chest or the front end.

Cab Fittings and Accessories.

While the designer of a locomotive must be familiar with, and provide for, the various cab fittings and accessories that are to be applied to the machine, he has nothing to do with their designing, and frequently is not even consulted regarding the selection of fittings from those that are upon the market. The manufacturing of these parts is in the hands of specialists, and the purchaser usually selects what he pleases and orders them put upon the engine. The part played by the designer in the matter is that of so locating them that they may be readily manipulated and their indications easily seen by the engineer and fireman.

These parts consist of the injector, engineer's valve, water gages, steam gage, whistle, lubricator, injector cock, cylinder cocks, blow-off cock, headlight, air-pump, and sander. It is essential that some of

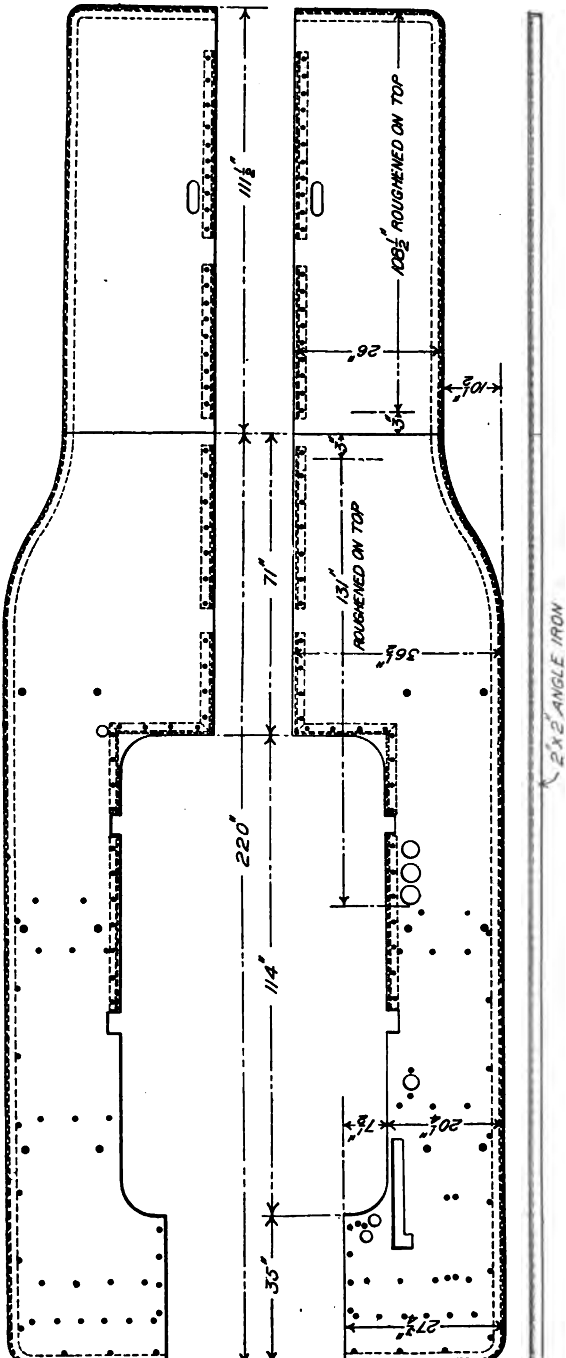

Fig. 16. Design of Running Boards.

these should be so placed as to afford the maximum of convenience to the engineer in the running of the engine. The reverse and throttle levers are the essential features of the engine itself, and near them must be located the handles for the working of the injector and the engineer's valve. One is needed in order to control the supply of water to the boiler, and the other for the manipulation of the air-brakes, for very few locomotives are now built that are not equipped with such brakes. With these handles easy of access, the engineer can control his train with safety, and the time expended in making the various movements is reduced to a minimum. Next to the injector and the engineer's valve come the whistle and water gages. The whistle lever is usually so located that it can be worked by reaching up and grasping either the lever itself or a cord attached to it that is near the roof; and the gage cocks are close at hand on the left, often near the engineer's valve. Owing to the introduction of the water glass, many engineers have come to depend upon it entirely for water level indications, with the result that it often happens that the gage cocks are placed out of reach. This should never be done for they should always be close at hand.

The steam and air gages are usually placed upon the top of the boiler and face directly to the rear. The air gage indicates the amount of air carried in the reservoir and train pipe, and should often be consulted by the engineer, especially after an application of the brakes, in order to note whether the pressures have been unduly reduced. So with the steam gage, he watches it when the pressure is low rather than when it is high, but for the fireman its visibility is of the first importance, and it should be so placed that it can be read from all points on the foot-plate and the tender, as well as from the seat upon the left-hand side. As these attachments are in constant use, and as it is upon them that the safe running of the train depends, the greatest pains should be exercised that they are so placed as to afford the utmost of convenience to the men. Of secondary importance are the handles for operating the cylinder cocks and sander. Usually it is necessary for the engineman to reach forward to the front of the cab or down to the floor to grasp them. Of the two, the sand lever had better be the more easy of access.

Within the cab there are a number of other attachments. The lubricator for oiling the valves and cylinders is usually placed upon the top of the boiler. As it works continuously after being started, it is never allowed to crowd more important parts out of desirable places. Under present conditions the lubricator should have at least three points of feeding, one for each steam chest and cylinder, and one for the air-pump. Sometimes a lubricator is used with a multiple feed with pipes leading to the several axle journals. The same requirement holds true for the location of the valves controlling the steam heat and operating the blower. They are only manipulated occasionally and so need not be in prominent positions; but, at the same time, care should be taken to put them where they will be readily accessible, and

not tuck them away where there is a likelihood of burning the hand when an attempt is made to reach them.

For fixtures that are applied outside the cab, the designer should see that they are properly located. The injector check, for example, should be placed somewhat back from the front tube-sheet, though there is by no means an agreement as to exactly where this should be. Practice varies in this respect from 10 inches to 5 feet back of the sheet and the probability is that 3 feet would be a good safe average location. This refers to the usual practice of placing the check on the side of the boiler and about on a line with its center. When this is done there should be an ample space of not less than 6 inches left between the opening and the first tube opposite it. When the injector check is placed in the back head, there should be a pipe leading from the inside opening forward, beneath the water line to a point within 3 feet or 4 feet of the front tube-sheet.

The headlight must be of such a size and so located that it will not interfere with the draft of the stack. The top of the stack is usually the highest point on the engine, so that ordinarily there is no trouble. Occasionally, however, the top of the ventilator of the headlight has been brought up flush with, or raised slightly above, the top of the stack. In these cases, when steam is shut off and the engine is drifting, an eddy is formed over the top of the headlight that turns down into the stack and drives the smoke and gases out into the cab.

All oil cups that are not forged solid with the parts that they are intended to serve should be of the strongest and most substantial construction; else they will become loosened and lost or broken by the jarring of the engine or the movement of the parts.

As already stated, the designer has little to say regarding the details of the cab fixtures that are to be used. A certain standard has been adopted by the customer or user, and this is specified. If the designing is done by the engineering department of the road that is to use the locomotive, it naturally follows that the regular standards will be applied. This applies not only to those items mentioned above but to other details as well, such as lagging for boiler and cylinders, the material for castings, the finish of the running-board, the type of air brake and pump, the size of the air reservoir, the arrangement of the air-pump exhaust, the whistle signal, gong, turret or combination fitting on the back end of the boiler, washout plugs, feed hose connections, lazy cocks, safety valves, vacuum valves, gage lamps, signal flags and lanterns, oil cans, jacks and engine tools, the color and method of painting, and frequently the material used. In all this the designer is merely a passive instrument who must place each individual piece in its proper position, see that provision is made for fastening it securely, and that it is readily accessible for manipulation and repairs. This last item is of the utmost importance and it cannot be emphasized too strongly. While a locomotive is intended for service on the road and nothing should be sacrificed to make it as efficient as possible, it must always be borne in mind that the working parts

will wear out and will eventually need to be repaired. For that reason all parts should, as far as possible, be so constructed and placed that they can be readily removed and replaced. Neglect of this precaution will result in that minor and general repairs will be far more expensive than they would have been had the parts been more accessible. This accessibility is also of the first importance in the matter of inspection. Where inspection is easy, it is much more apt to be thorough than where it is difficult, and neglect along these lines finds its direct reflection in the cost for repairs. Finally, strength and security should invariably occupy the first place in the mind of the designer, and cheapness of first cost should not blind him to the requisities of strength and durability. And as he takes particular pains that the frames, boiler and cylinders are rigidly fastened in position, so he should see to it that all of the accessories should be secure, using flanges for all larger boiler connections instead of screwed connections, for it will be upon the detailed attention that has been paid to these seemingly minor matters that the successful operation of the locomotive may depend.

CHAPTER IV.

THE TENDER.

The locomotive having been designed for the service for which it is intended, it remains to provide it with a suitable tender for carrying the supply of fuel and water. In American practice this tender, until recently, almost invariably consisted of a U-shaped tank carried on a metal framework, though wood is still occasionally used on the smaller engines for this purpose. The tank, in turn, is mounted upon two four-wheeled bogie trucks. The fuel is carried between the legs of the U of the tank.

Tank Capacity.

The first point to be settled in the design is the capacity of the tank. This will depend not only upon the size of the engine and the work which it is to do, as governing its steam consumption, but also upon the location of the water tanks and the distances that will have to be run between stops. Where the tender is to be fitted with a water scoop for taking water from track tanks, and these are located at frequent intervals, obviously the tender can be of less capacity then where long runs must be made. Further, a variation in tank capacity will be called for between a mountain and level division with the same distance between water stops, because of the greater steam consumption upon adverse grades when compared with the level. As a matter of fact, however, no actual distinction is made in this respect between engines on the same system. Provision is made for a sufficient water supply for the heaviest work that is to be done on any part cf the line; and, then, if the engine is transferred to lighter work there is simply an excess. The one thing needful is that the tank should hold enough water to do a little more than supply the engine, when working at full capacity over the severest section of the line.

Tank Construction.

There are two general types of tank construction that are used in the United States, known as the plain U tank and the water bottom type. The latter type varies somewhat and generally has no water legs, but in their stead a slope from the back end of the coal space, extending down toward the front and reaching across the tank, and then turning horizontally to form a water space beneath the floor. It is used where the foot-board of the engine is high, so as to raise the tank floor up to the same level for the convenience of the fireman.

Figs. 17 and 18 illustrate the tank designed for use on the consolidation and Atlantic locomotives, respectively. The tank for the consolidation locomotive has a capacity of 6,000 gallons and is without a water bottom. It consists of a U-shaped tank, with a sharp incline at the

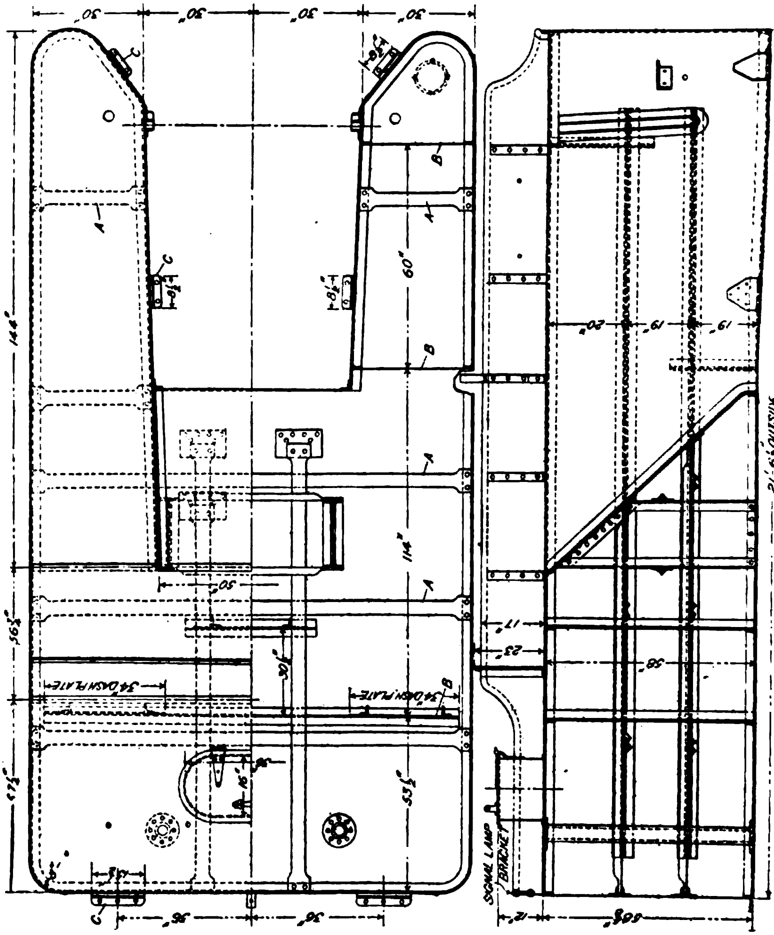

Fig. 17. Tank of 6,000 Gallons Capacity for Consolidation Locomotive.

back of the coal space, and is bolted to the tender frame, upon which it rests. The outsides are vertical and are made of 3/16-inch to 1/4-inch steel. The framing for the tank is on angles, usually 2½ inches by 2½ inches, arranged on the inside, and it is to them that the plates are riveted. Ordinarily, ½-inch rivets are used, spaced 1½ inch between centers. In the construction of the tank it is unnecessary to calk the seams, as in the case of a boiler, because there is no internal pressure other than that due to the weight of the water. For that reason the joint between the plate and the angle is laid with a strip of heavy paper (usually tarred) between, so that, when the rivet is driven, the two parts are drawn together to form a water-tight joint. While there are no pressures against the sides that can cause trouble, the tank itself, as a whole, is subjected to heavy strains, and it must

be braced accordingly. These strains arise from the surging of the contained water due to the varying motions of the tender. Whenever the brakes are applied, the water rushes to the front and banks up in the legs, and then flows back to the rear. In addition to this, there is a constant surge to and fro laterally as the frame rolls cn the trucks. When it is taken into consideration that the water in a half-empty tank weighs from 24,000 to 32,000 pounds, it will readily be understood that the effect of the wave action of this weight, as it moves about, may be very severe. There is a two-fold tendency in this action. One is to shift the tank upon the frame, and the other is to cause the sides to bulge. The surging effect is lessened by the interposition of splash plates in the legs and body, by which the space through which the water must flow is greatly contracted, and the effects of a high momentum destroyed. For the protection of the side plates against bulging, they are stayed with internal transverse; vertical and horizontal bracings of plates and angles riveted to the sheets so as to effect the most uniform distribution of the stresses possible, and so prevent any deformation of the tank.

The splash or dash plates are riveted to the vertical angles as shown at *B* in Fig. 17. In spite of these precautions the splashing effect of the water longitudinally is still very severe, so that the grill of internal stiffening angles should be such that the openings should not be more than from 18 inches to 24 inches across. The tank, as a whole, must be so firmly fastened to the frame that there is no possible danger of displacement because of the surging of the water or the shocks to which it will be subjected in service. This fastening is usually effected by means of heavy angle lugs having one leg riveted to the vertical plates and the other bolted to the framing. Although it is the side or vertical portion of the tank that is called upon to withstand the heaviest shccks due to the movement of the water, the top and bottom sheets, by presenting the larger surfaces to coal and the weather, are more expcsed to wear and corrosion, and are, therefore, generally made from ¼ inch to 5/16 inch thick. It is especially desirable that the face of the coal space should be as smooth as possible, so as to afford no lodging place for coal or dirt, as such accumulations are apt to promote corrosion.

At the front end of the coal space, vertical angles are riveted to the legs to hold the coal boards, and wherever holes are cut for pipe or other connections the sheet should be stiffened by a plate of ample thickness and size, usually in the form of a flange riveted over the place. It is impossible to give any formula for calculating the stresses to which a tender tank may be subjected, because they are so varied in character and so irregular in application that it is not known what they are. For that reason it would be well to follow the general features that have been set forth in the designing of the tank, both as to method of construction and the dimensions of the material, although with the consideration that a reasonable increase may be made in the case of tanks of enlarged capacity.

In addition to the arrangements that have to be made to enable the tank to carry its load successfully, there are other attachments, to the location of which careful attention should be paid. This refers to such items as a shield that is placed across the top of the tank, back of the coal space, to protect the tank opening from flying pieces of coal and debris; it is made of wood or steel plates from 20 inches to 36 inches in height. The tank opening may be either round or oblong, and if oblong, the long diameter usually extends across the tank, as shown in the cuts Figs. 17 and 18. Sometimes, however, it extends lengthwise. The effect is the same in either case—the elongated opening avciding the necessity for a close spotting at water stops.

There are handholds, or grab irons, to be placed near the steps at the front end, or in accordance with the legal requirements governing safety appliances. The lantern brackets at the rear are usually mere hooks located to suit the requirements of the purchaser. The same thing holds true regarding the tool-boxes. Usually there is a box on top of the forward end of each leg of the tank, provided with locks and keys, and in which the lighter engine supplies are kept, such as oil, waste, and hand tools; while, at the rear, either on top of the tank or behind it on the frame, there is another, extending across the engine, in which heavy supplies, such as chain, jacks, extra coupler knuckles and the like, are stored. These boxes are heavy and made of wood, and must be securely bolted in position.

The other form of tank, that with a water bottom extending forward beneath the floor space, is shown in the engraving of the tank of the Atlantic locomotive. In this case it is given a capacity of 8,000 gallons, and is correspondingly longer and deeper than the one of a smaller capacity used with the consolidation engine. In other respects it rescmbles in construction the one already described. There are, of course, some variations in detail, as, for example, the ladder at the back, which is required because of the great depth (68 inches), making the top so high that a man cannot readily raise himself up to it, as is possible in the case of the smaller tank. It will be noticed, too, that cupboards with doors are placed in the forward end of each leg. These are points that the designer must be prepared to think of and provide, if they are called for in the specifications.

In these large tanks the apron for holding the coal in place is usually made vertical and flush with the side, instead of being flared, as is the practice on the older and smaller tenders, and is only carried back as far as the manhole shield, the space about the hole being protected by a low railing. The top edge of the apron is always stiffened, usually by a bar of half-round iron riveted on the upper edge. These are minor, but somewhat important, details to which a designer should pay close attention, that the effect may be satisfactory not only to the eye, but in the service which is to be rendered.

Tender Frame Construction.

The frame upon which the tank rests was formerly made of wood, but present construction is usually of steel. This frame is usually a

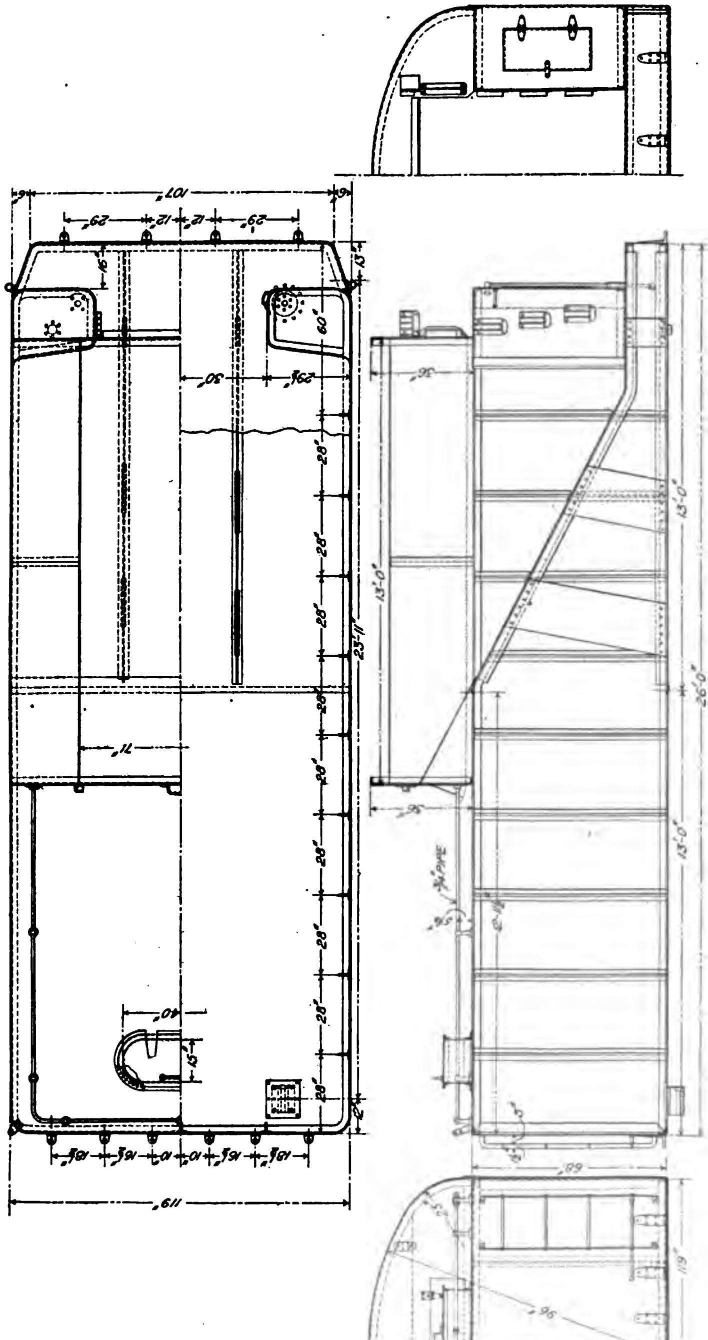

Fig. 18. Tank of 8,000 Gallons Capacity for Locomotive of the Atlantic Type.

short rectangular structure built of rolled sections firmly bolted and riveted together, and so braced diagonally that it cannot be twisted or distorted by any of the ordinary stresses to which it may be subjected. The load imposed is always uniformly distributed over the whole length. In this connection, the under frame should be of ample strength to carry the whole load, as the water and coal rest fully on the lower frame and are supported to only a slight extent by the vertical walls of the tank. However, the distance between points of support of the frame is short, running from 10 feet to 12 or 13 feet, which is the distance between truck centers.

At the point of support the bolsters should be of ample strength and of great stiffness. Any flexibility or limberness will be sure to manifest itself in the imposition of an excessive weight on the side bearings and a racking of the structure as a whole. Therefore, the body bolsters should be so designed that twice the maximum load can be sustained on the center-plate, with an ample margin to spare in the way of a factor of safety. It is not known exactly what the actual stresses imposed upon tender bolsters may be, other than that they are, at times, at least 50 per cent in excess of the static loads.

The frame illustrated in Fig. 19 is that intended for use under the short 6,000-gallon tank for the consolidation locomotive. The coal, water, accessories, and tank will weigh, approximately, 80,000 pounds, to which is added the weight of the frame, which will run the static load on the bolsters up to about 90,000 pounds, or 45,000 pounds on each one. Therefore, each bolster should be of ample strength to sustain at least 68,000 pounds. In this case, four 10-inch channels are used as the sills, and the span between center-plates is 11 feet 8 inches, with an overhang of something less than half the amount. Each of these channels is capable of supporting a uniformly distributed load of about 19 450 pounds for the span given, or 77,800 pounds for the four sills, with a fiber stress of 16,000 pounds per square inch. As this span is but 55 per cent of the total length of the frame, the load actually imposed will be about 49,500 pounds, which, as compared with the 77,800 pounds, cuts down the fiber stress correspondingly; so that, as far as the longitudinal members of the structure are concerned, there is ample strength to carry the load and keep well within allowable fiber stresses.

The perfect uniformity of the loading makes it possible to determine the probable distribution of the same on the bolsters more accurate than is possible on cars, where eccentric and local loading is apt to occur. Where it is intended that the side bearings shall be in contact and carry a due proportion of the load, it is not necessary that the bolster shall be as stiff as where the whole load is to be on the center-plate. In this case, the former condition prevails and the bolster consists essentially of two broad plates 28 inches wide riveted to the top and bottom flanges of the sills, and extending across the full width of the frame. They are ½ inch thick. A diagonal plate of the same width is placed between the center and side sills, and a filling

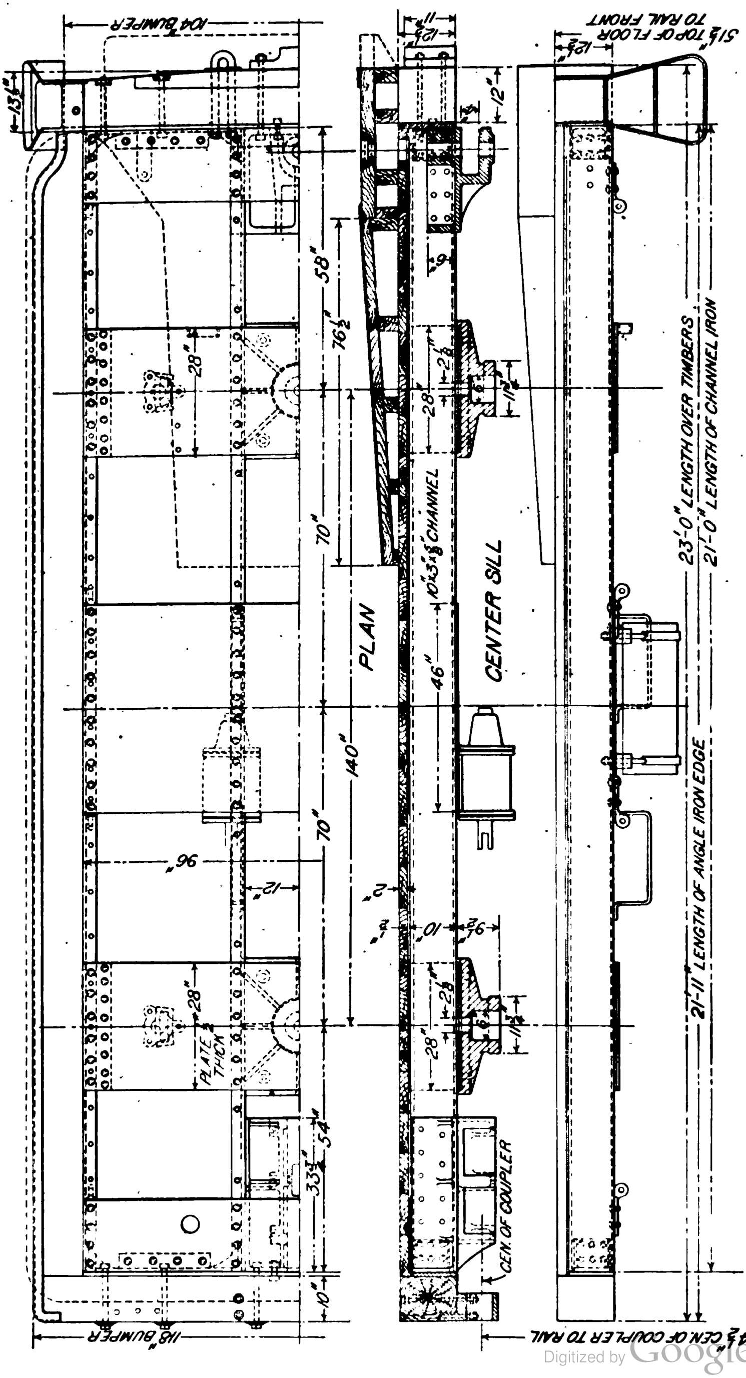

Fig. 19. Tender Frame for Locomotive of the Consolidation Type.

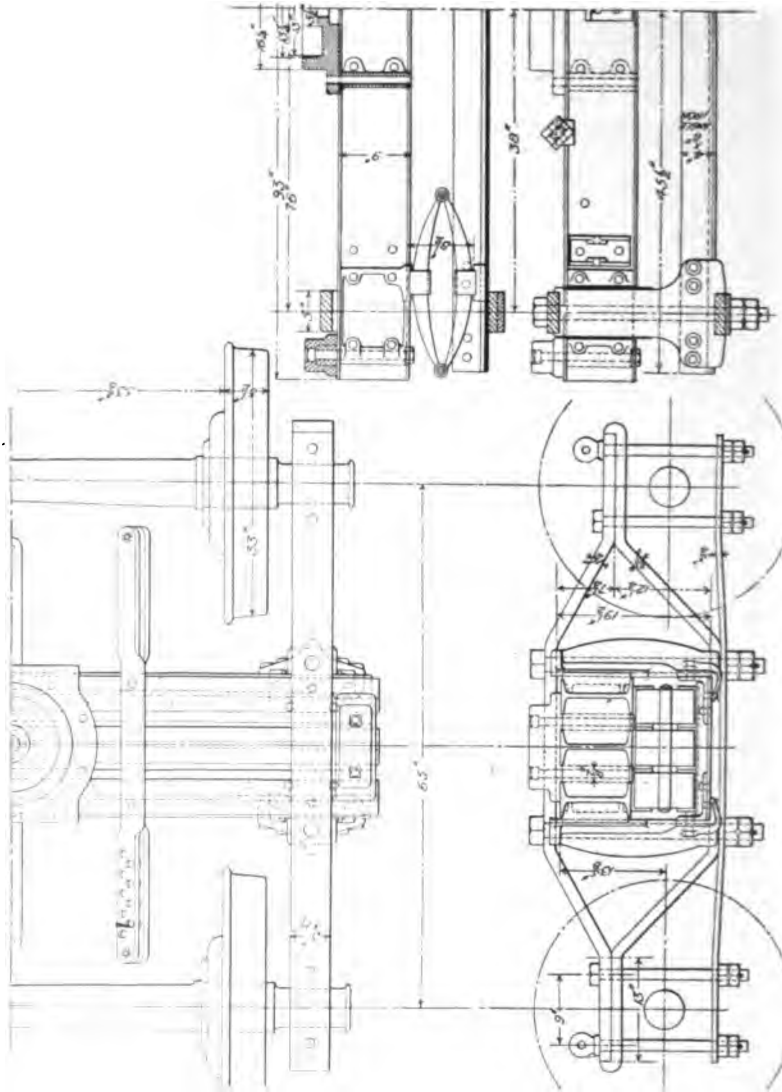

Fig. 20. Atlantic Locomotive Tender Truck for Tank of 8,000 Gallons Capacity.

piece is inserted above the side bearings. In this way the plates are protected from bending due to the load on the side sill, and a portion of the load is carried down to the center-plate.

The center-plate should have an ample bearing surface to carry the load, and should have sufficient depth to receive the male portion, so that derailment or excessive shocks should not be able to cause a separation of the parts. As to what constitutes an ample bearing surface, there is no agreement of practice or opinion. The loads in use

vary from 300 pounds to 3,000 pounds per square inch of area in contact. In 1903 the Master Car Builders' Association adopted a standard form of center-plate having 100 square inches area. This, for a loaded car, would give a pressure of about 650 pounds per square inch. In the center-plate of the tender under consideration, the area of contact is 80.16 square inches, and this, with a load of 45,000 pounds, has an imposed pressure of something more than 560 pounds per square inch, which is well within the limits of the Master Car Builders' recommendations. The form of the center-plate depends upon the adjoining parts, but in any case it should be heavy and strongly ribbed to avoid deflection, and is usually made of cast iron.

Ranking close in importance with the elements forming the frame are the castings at the front and back to which the draft and buffing rigging is attached. These castings are of cast iron or steel, and are of heavy sections, securely riveted between the center sills. There is no rule for the calculation of the stresses to which they may be subjected, as the tractive effort of the engine is often many times exceeded. The draw-pin, at the forward end, should be at least 3 inches in diameter, and the casting must be of such proportions as to resist the full shearing strength of such a pin. The end plates may be 1 inch thick, riveted across the ends of the sills and held in place by suitable angles.

Tender Trucks.

The trucks used under tenders are of great variety of form. Many of these forms are patented or the objects of special manufacture. When such a truck is to be used, the engine designer has merely to specify the weights that are to be carried, and then provide for connection to the form and dimensions offered. No attempt will be made to describe a variety of trucks, but attention will be confined to the one using an ordinary diamond side frame, like that extensively used under freight cars. As the duties of a tender are similar to those of a freight car carrying heavy bulk freight, it is quite necessary that the trucks used should be similar. Accordingly, the wheel-base is from 5 feet to about 5 feet 6 inches. The side frames, too, are identical in construction with those used under freight cars, the bars being proportioned to the load to be carried, with an allowance for extra heavy service due to the location of the tender immediately behind the engine.

The simplicity of the form of the diamond truck would render the stresses, due to vertical load, as they fall upon the arch bars, easy of analysis, were it not for the blows and horizontal thrusts to which they are subjected. Take the truck for the consolidation locomotive, for example; it is calculated to carry a maximum load of about 49,000 pounds. Of this, one-half, or 24,500 pounds, is upon each side-frame. The bolts are called upon to sustain but comparatively little in tensional stress, but must be able to withstand the shear put upon them in giving rigidity to the truss which is formed by the various tension and compression members of which it is composed. By plotting the frame and solving by a simple parallelogram of forces, it

will be found that the lower arch bar would have to carry a tension stress of about 28,000 pounds, while the stress on the upper one would be that of compression and be about 27,700 pounds, provided the bars were straight between the points of support. Owing to the requirements of construction, the bars have two bends between the points of support, throwing them out of a direct line, and introducing a bending moment that will be likely to crack the lower bar at the upper face of the lower bend. For this reason the bars must be made considerably stronger than would be required were there no such bending effect. The amount of this will depend upon the distance of the center of the bend from the point of support, and the angle at which the bar stands in the frame. The fiber stress set up by these conditions can be approximately determined by the use of the same methods as those suggested for the determination of the general stresses set up in the frame by the vertical loading. In addition to the vertical loads and the stresses resulting therefrom, there are severe lateral stresses set up by the lurching of the tender from side to side and the effect of centrifugal force on curves. These last are calculable by the regular formula for centrifugal action, but for the other stresses, due to the unevenness of the track, cramping of the wheels, high speed and derailments, there are no data available, and recourse must be had to the experience of the past.

Turning to the two trucks shown, that for the consolidation locomotive, Fig. 21, has been given a lower arch bar of 1⅝ inch thickness and an upper one of 1½ inch, and a tie bar of ¾ inch, all having a width of 4½ inches, based upon the considerations given above. The lower bars act in tension and the upper in compression. The bolster is of cast steel, and the axles are those adopted as the standard by the Master Car Builders' Association for cars of 80,000 pounds capacity.

In the truck for the tender of the Atlantic locomotive, Fig. 20, the wheel-base is greater. The heavier loads and harder service which the high speed entails, put greater stresses on the frames, which must, therefore, be made heavier, and with this an increased diameter of column bolts are used, which are made 1¾ inch diameter. This calls for a correspondingly larger hole, so that the width of the bars is made 5 inches, and this is in accordance with the greater width of the journal, which is that of the Master Car Builders' standard for cars of 100,000 pounds capacity. The top arch-bar has, in this case, a thickness of 1½ inch, as before, while that of the lower one, owing to its greater width, is made 1¼ inch, while the bottom or tie bar is ¾ inch thick.

The bolster of the Atlantic engine consists of three 9-inch I-beams held by suitable separating pieces to the columns and frames.

With this, the most important points involved in the designing of a locomotive have been treated, although there are numerous details that have been left untouched, since they belong to the class of detail work or of special manufacture. The object of this work is merely to present a guide to the general work, for it will be readily appreci-

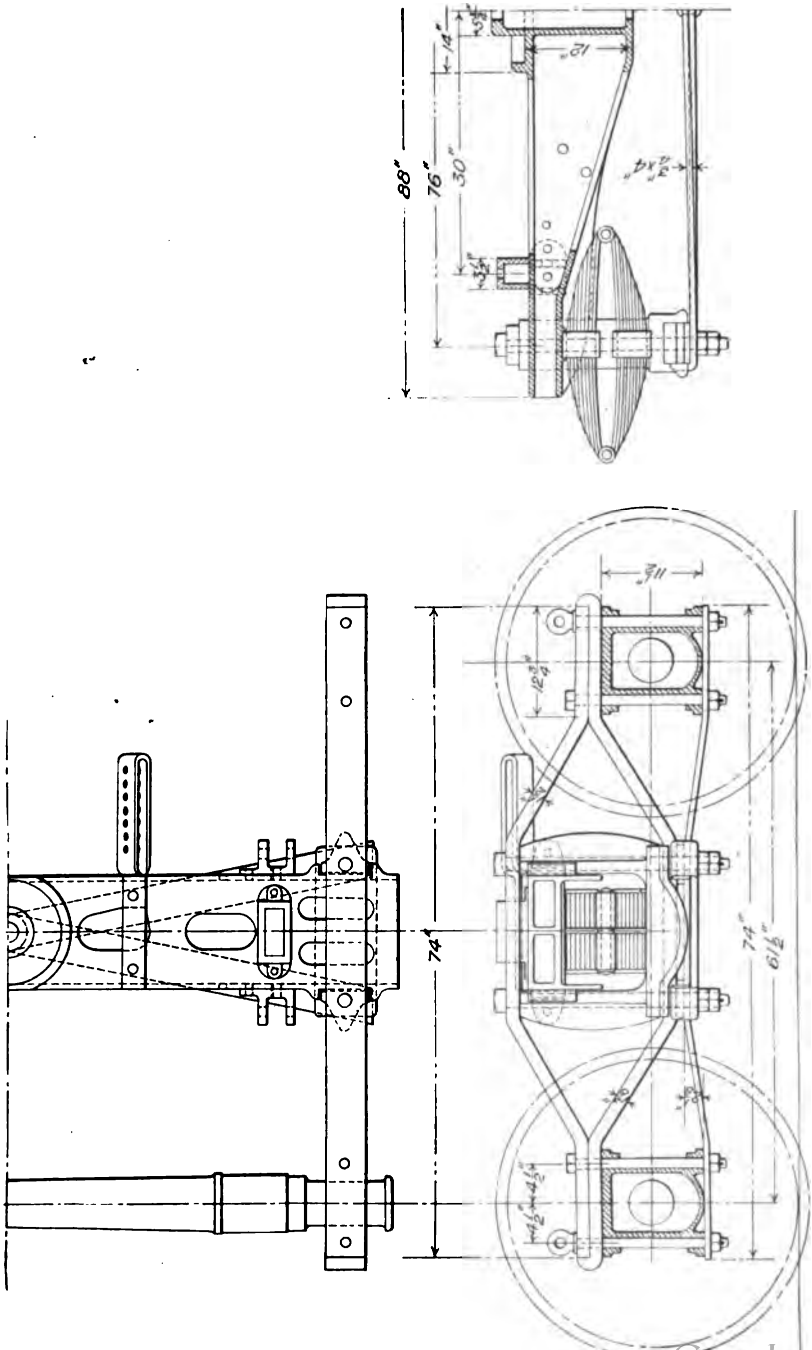

Fig. 21. Consolidation Locomotive Tender Truck for Tank of 6,000 Gallons Capacity.

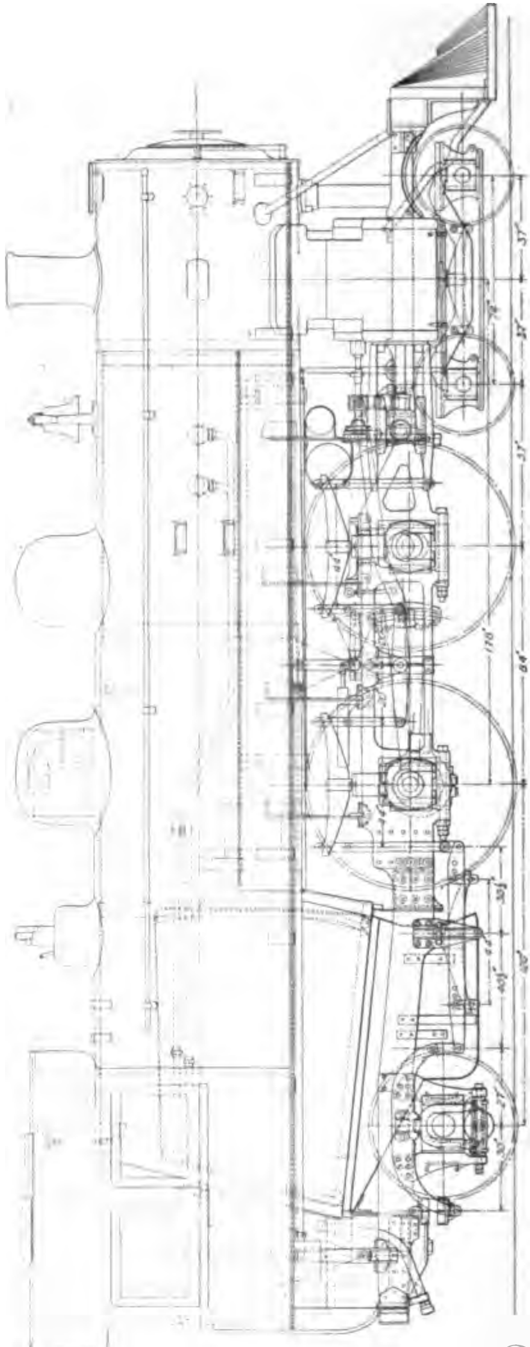

Fig. 22. Locomotive of the Atlantic Type.

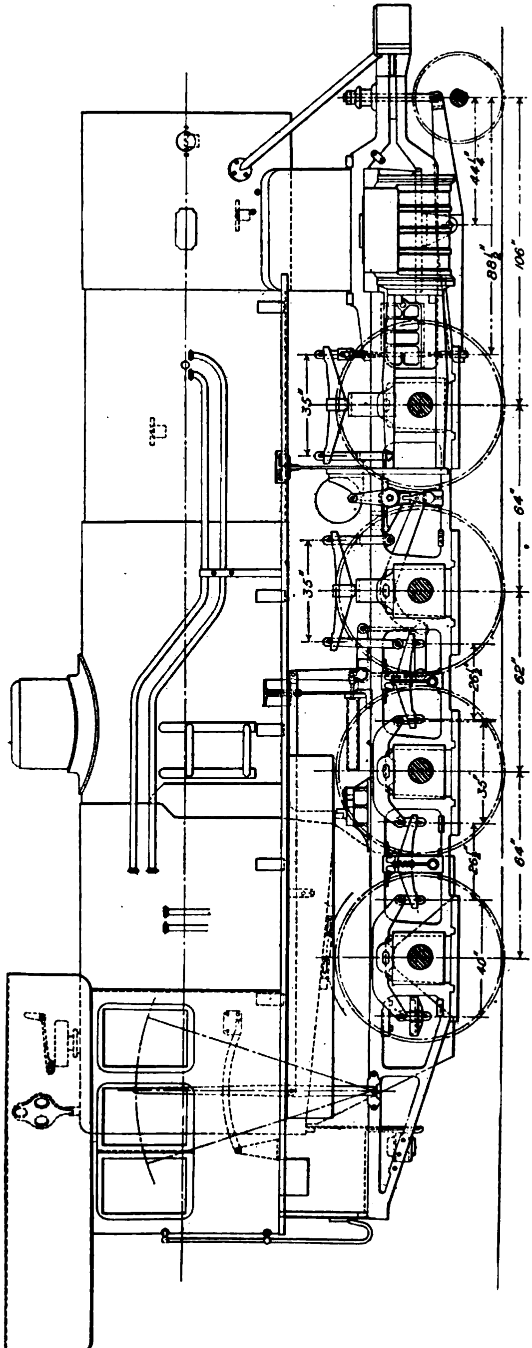

Fig. 28. Locomotive of the Consolidation Type.

ated, that anyone, to undertake the designing of a locomotive, must first have had considerable drawing office experience, to whom the filling out of the parts omitted in this work will be comparatively simple. The result of what has been done is shown in Figs. 22 and 23.

In conclusion, there are a few suggestions that should be constantly kept in mind during the whole progress of the work, whether it be that of designing a locomotive, a stationary engine or any other piece of machinery, and that is to bring the three elements of utility, simplicity and beauty into one harmonious whole. They are the three important factors entering into the composition, and no one of them should ever be disregarded, for they can always be combined without additional expense, and this combination should be made to enter into every detail of the work.

A successful designer must, therefore, be not only a master capable of solving the technical problems involved in a manner to obtain the highest degree of efficiency, but he must have that practical knowledge gained from experience, from which he will be able to choose the simplest methods of construction, and to this should be added an inborn instinct as to the fitness of things, which should have been cultivated by practice and study, whereby the results, though, perhaps, not falling quite within the recognized realms of art, should still be of such a character that they are pleasing to the eye, and as such claim the attention as artistic creations.

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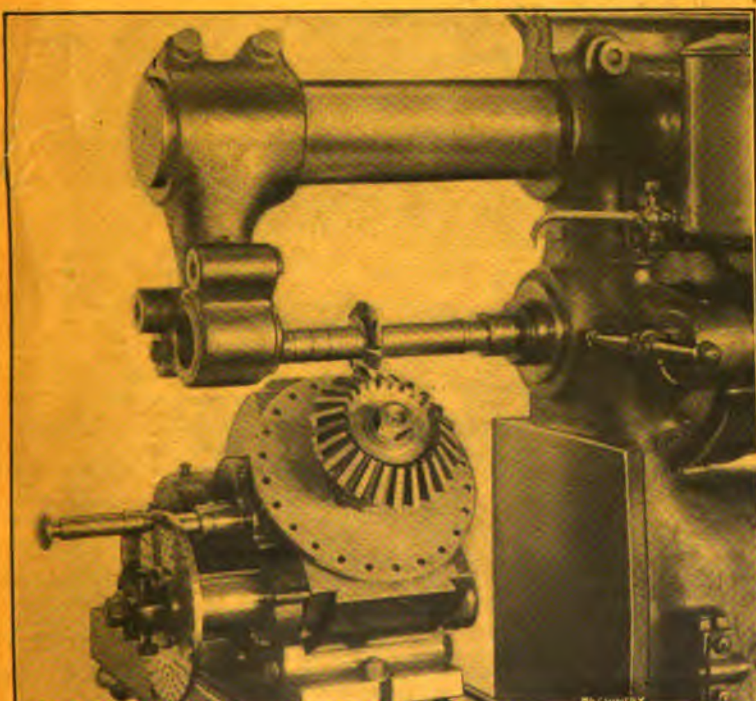

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BEVEL GEARING

By Ralph E. Flanders

Third Revised Edition

CONTENTS

CHAPTER I

BEVEL GEAR RULES AND FORMULAS

Bevel gearing, as every mechanic knows, is the form of gearing used for transmitting motion between shafts whose center lines intersect. The teeth of bevel gears are constructed on imaginary pitch cones in the same way that the teeth of spur gears are constructed on imaginary pitch cylinders. In Fig. 1 is shown a drawing of a pair of bevel gears of which the gear has twice as many teeth as the pinion. The latter thus revolves twice for every revolution of the gear. In Fig. 2 is shown (diagrammatically) a pair of conical pitch surfaces driving each other by frictional contact. The shafts are set at the same center angle with each other, as in Fig. 1, and the base diameter of the gear cone is twice that of the pinion cone, so that the latter will revolve twice to each revolution of the former. This being the case, the cones shown in Fig. 2 are the pitch cones of the gears shown in

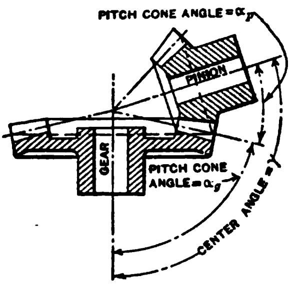


Fig. 1. Bevel Gear and Pinion

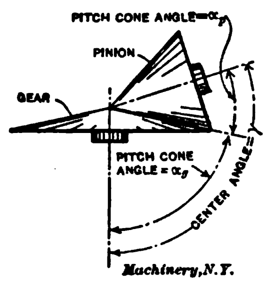


Fig. 2. Pitch Cones of Gears Shown in Fig. 1

Fig. 1. We may therefore define the term "pitch cone" as follows: The pitch cones of a pair of bevel gears are those cones which, when mounted on the shafts in place of the bevel gears, will drive each other by frictional contact in the same velocity ratio as given by the bevel gears themselves.

The pitch cones are defined by their pitch cone angles, as shown in Fig. 2. The sum of the two pitch cone angles equals the center angle, the latter being the angle made by the shafts with each other, measured on the side on which the contact between the cones takes place. The center angle and the pitch cone angles of the gear and the pinion are indicated in Fig. 1.

Different Kinds of Bevel Gears

In Fig. 3 is shown a pair of bevel gears in which the center angle (γ) equals 90 degrees, or in other words, the figure shows a case of right angle bevel gearing. To the special case shown in Fig. 4 in which the number of teeth in the two gears is the same, the term miter gearing is applied; here the pitch cone angle of each gear will always equal 45 degrees.

When the pitch cone angle is less than 90 degrees we have acute angle bevel gearing, as shown in Fig. 5. When the center angle is greater than 90 degrees, we have obtuse angle bevel gearing, shown in Fig. 6 and also in Fig. 1. Obtuse angle bevel gearing is met with occasionally in the two special forms shown in Figs. 7 and 8. When the pitch cone angle a_g equals 90 degrees, the gear g is called a crown gear. In this case the pitch cone evidently becomes a pitch plane, or

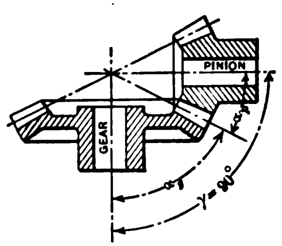


Fig. 3. Right Angle Bevel Gearing

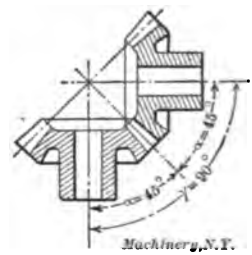


Fig. 4. Miter Gearing

disk. When the pitch cone angle of the gear is more than 90 degrees, as in Fig. 8, this member is called an internal bevel gear, and its pitch cone when drawn as for Fig. 2, would mesh with the pitch cone of the pinion on its internal conical surface. These two special forms of gears are of rare occurrence.

Bevel Gear Dimensions and Definitions*

In Fig. 9, which shows an axial section of a bevel gear, the pitch lines show the location of the periphery of the imaginary pitch cone.

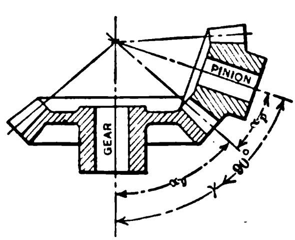


Fig. 5. Acute Angle Bevel Gearing

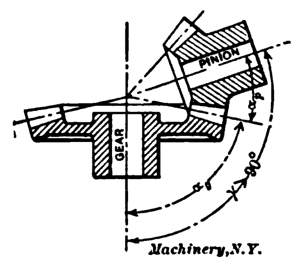


Fig. 6. Obtuse Angle Bevel Gearing

The pitch cone angle is the angle which the pitch line makes with the axis of the gear. The pitch diameter is measured across the gear drawing at the point where the pitch lines intersect the outer edge of the teeth. The teeth of bevel gears grow smaller as they approach the vertex *O* of the pitch cone, where they would disappear if the teeth were cut for the full length of the face. In speaking of the pitch of a bevel gear we always mean the pitch of the larger or outer ends of the teeth. Diametral and circular pitch have the same meaning as in the case of spur gears, the diametral pitch being the num-

*MACHINERY, February, 1910.

ber of teeth per inch of the pitch diameter, while the circular pitch is the distance from the center of one tooth to the center of the next, measured along the pitch diameter at the back faces of the teeth. The addendum is the height of the tooth above the pitch line at the large end. The dedendum (the depth of the tooth space below the pitch line) and the whole depth of the tooth are also measured at the large end.

The pitch cone radius is the distance measured on the pitch line from the vertex of the pitch cone to the outer edge of the teeth. The width of the face of the teeth, as shown in Fig. 9, is measured on a line parallel to the pitch line. The addendum, whole depth and thickness of the teeth at the small or inner end may be derived from the corresponding dimensions at the outer end, by calculations depending on the ratio of the width of face to the pitch cone radius. (See *s*, *w* and *t* in Fig. 12.)

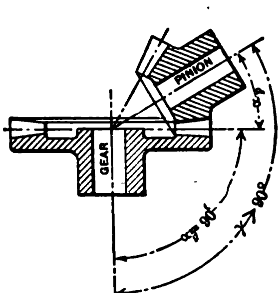


Fig. 7. Crown Gear and Pinion

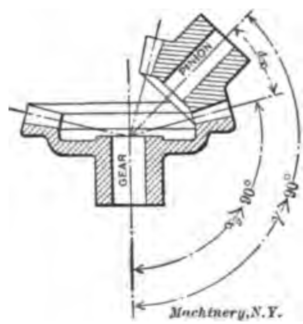


Fig. 8. Internal Bevel Gear and Pinion

The addendum angle is the angle between the top of the tooth and the pitch line. The dedendum angle is the angle between the bottom of the tooth space and the pitch line. The face angle is the angle between the top of the tooth and a perpendicular to the axis of the gear. The edge angle (which equals the pitch cone angle) is the angle between the outer edge and the perpendicular to the axis of the gear. The latter two angles are measured from the perpendicular instead of from the axis, for the convenience of the workman in making measurements with the protractor when turning the blanks. The cutting angle is the angle between the bottom of the tooth space and the axis of the gear.

The angular addendum is the height of tooth at the large end above the pitch diameter, measured in a direction perpendicular to the axis of the gear. The outside diameter is measured over the corners of the teeth at the large end. The vertex distance is the distance measured in the direction of the axis of the gear from the corner of the teeth at the large end to the vertex of the pitch cone. The vertex distance at the small end of the tooth is similarly measured.

The shape of the teeth of a bevel gear may be considered as being the same as for teeth in a spur gear of the same pitch and style of

tooth, having a radius equal to the distance from the pitch line at the back edge of the tooth to the axis of the gear, measured in a direction perpendicular to the pitch line. This distance is dimensioned $\frac{D'}{2}$ in Fig. 12. The number of teeth which such a spur gear would have, as determined by diameter *D'* thus obtained, may be called the "number of teeth in equivalent spur gear," and is used in selecting the cutter for forming the teeth of bevel gears by the formed cutter process.

In two special forms of gears, the crown gear, Fig. 10, and the internal bevel gear, Fig. 11, the same dimensions and definitions apply as in regular bevel gears, though in a modified form in some cases. In the crown gear, for instance, the pitch diameter and the outside diameter are the same, and the pitch cone radius is equal to ½ the pitch diameter. The addendum angle and the face angle are also the same. The angular addendum becomes zero, and the vertex distance is equal to the addendum. The number of teeth in the equivalent spur gear becomes infinite, or in other words, the teeth are shaped like those of a rack.

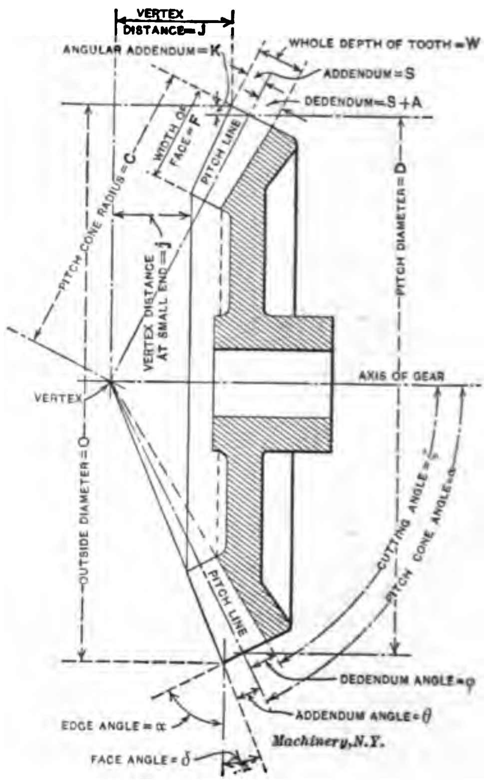


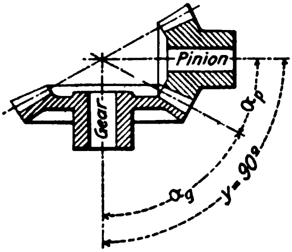
Fig. 9. Dimensions, Definitions and Reference Letters for Ordinary Bevel Gear

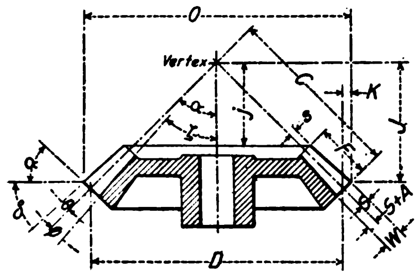
When the pitch cone angle is greater than 90 degrees, so that the gear becomes an internal bevel gear, as in Fig. 11, the outside diameter (or edge diameter as it is better called in the case of internal gears) becomes less than the pitch diameter. Otherwise the conditions are the same although many of the dimensions are reversed in direction.

Rules and formulas for calculating the dimensions of bevel gears are given on pages 7, 9, 11, and 13. The following reference letters are used:

CHART FOR SOLUTION OF BEVEL GEAR PROBLEMS.—I

Bevel Gears with Shafts at Right Angles.





Note · α_p = Pitch Cone Angle of Pinion
α_g = Pitch Cone Angle of Gear
N_p = Number of Teeth in Pinion, etc.

Use Rules and Formulas 1-21 in the order given.

No.		To Find	Rule	Formula
1		Pitch Cone Angle (or Edge Angle) of Pinion	Divide the number of teeth in the pinion by the number of teeth in the gear to get the tangent	$\tan \alpha_p = \frac{N_p}{N_g}$
2		Pitch Cone Angle (or Edge Angle) of Gear	Divide the number of teeth in the gear by the number of teeth in the pinion to get the tangent	$\tan \alpha_g = \frac{N_g}{N_p}$
3		Proof of Calculations for Pitch Cone Angles	The sum of the pitch cone angles of the pinion and gear equals 90 degrees	$\alpha_p + \alpha_g = 90°$
4		Pitch Diameter	Divide the number of teeth by the diametral pitch; or multiply the number of teeth by the circular pitch and divide by 3.1416	$D = \frac{N}{P} = \frac{NP'}{\pi}$
5	These dimensions are the same for both gear and pinion	Addendum	Divide 1.0 by the diametral pitch; or multiply the circular pitch by 0.318	$S = \frac{1.0}{P} = 0.318P'$
6		Dedendum	Divide 1.157 by the diametral pitch; or multiply the circular pitch by 0.368	$S+A = \frac{1.157}{P} = 0.368P'$
7		Whole Depth of Tooth Space	Divide 2.157 by the diametral pitch; or multiply the circular pitch by 0.687	$W = \frac{2.157}{P} = 0.687P'$
8		Thickness of Tooth at Pitch Line	Divide 1.571 by the diametral pitch; or divide the circular pitch by 2	$T = \frac{1.571}{P} = \frac{P'}{2}$
9		Pitch Cone Radius	Divide the pitch diameter by twice the sine of the pitch cone angle	$C = \frac{D}{2 \times \sin \alpha}$
10		Addendum at Small End of Tooth	Subtract the width of face from the pitch cone radius, divide the remainder by the pitch cone radius and multiply by the addendum	$s = S \times \frac{C-F}{C}$
11		Thickness of Tooth at Pitch Line at Small End	Subtract the width of face from the pitch cone radius, divide the remainder by the pitch cone radius and multiply by the thickness of the tooth at the pitch line	$t = T \times \frac{C-F}{C}$
12		Addendum Angle	Divide the addendum by the pitch cone radius to get the tangent	$\tan \theta = \frac{S}{C}$
13		Dedendum Angle	Divide the dedendum by the pitch cone radius to get the tangent	$\tan \phi = \frac{S+A}{C}$

N = number of teeth,
P = diametral pitch,
P' = circular pitch,
π = 3.1416, (pi),
α = pitch cone angle and edge angle, (alpha),
γ = center angle, (gamma),
D = pitch diameter,
S = addendum,
$S + A$ = dedendum (A = clearance),
W = whole depth of tooth space,

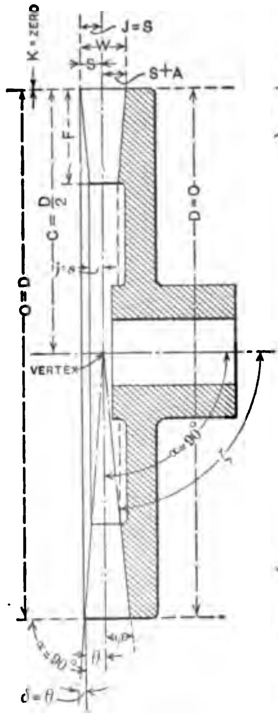


Fig. 10. Dimensions for Crown Gear

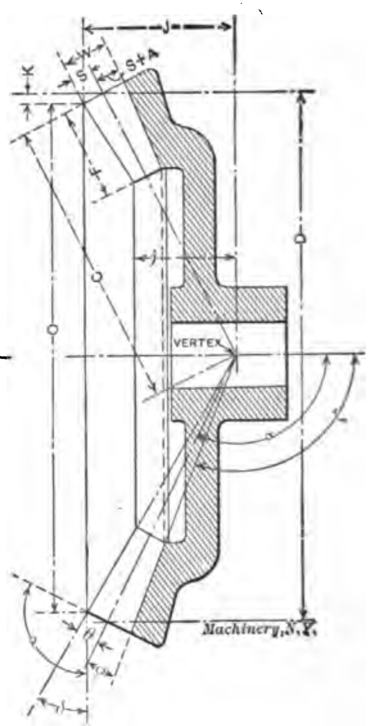


Fig. 11. Dimensions for Internal Bevel Gear

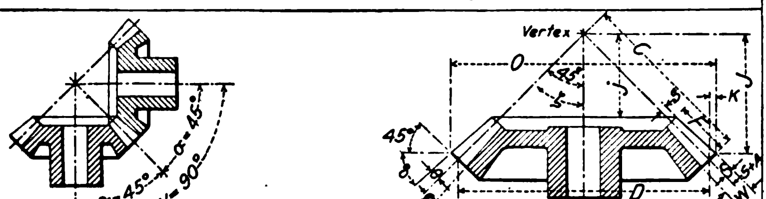
T = thickness of tooth at pitch line,
C = pitch cone radius,
F = width of face;
s = addendum at small end of tooth,
t = thickness of tooth at pitch line at small end,
θ = addendum angle, (theta),
ϕ = dedendum angle, (phi),
δ = face angle, (delta),
ζ = cutting angle, (zeta),
K = angular addendum,

CHART FOR SOLUTION OF BEVEL GEAR PROBLEMS.—II

Bevel Gears with Shafts at Right Angles (Continued).

No.	To Find	Rule	Formula
14	Face Angle	Subtract the sum of the pitch cone and addendum angles from 90 degrees	$\delta = 90° - (\alpha + \theta)$
15	Cutting Angle *	Subtract the dedendum angle from the pitch cone angle	$\zeta = \alpha - \varphi$
16	Angular Addendum	Multiply the addendum by the cosine of the pitch cone angle	$K = S \times \cos\alpha$
17	Outside Diameter	Add twice the angular addendum to the pitch diameter	$O = D + 2K$
18	Apex Distance	Multiply one-half the outside diameter by the tangent of the face angle	$J = \frac{O}{2} \times \tan\delta$
19	Apex Distance at Small End of Tooth	Subtract the width of face from the pitch cone radius; divide the remainder by the pitch cone radius and multiply by the apex distance	$j = J \times \frac{C-F}{C}$
20	Number of Teeth in Equivalent Spur Gear	Divide the number of teeth by the cosine of the pitch cone angle	$N' = \frac{N}{\cos\alpha}$
21	Proof of Calculations by Rules Nos. 9, 12, 14, 16 and 17	The outside diameter equals twice the pitch cone radius multiplied by the cosine of the face angle and divided by the cosine of the addendum angle	$O = \frac{2C \times \cos\delta}{\cos\theta}$

Mitre Bevel Gearing.



Use Rules and Formulas 22, 4-8, 23, 10-13, 24-26, 17-19, 27 and 21 in the order given. All dimensions thus obtained are the same for both gears of a pair

No.	To Find	Rule	Formula
22	Pitch Cone Angle	Pitch cone angle equals 45 degrees	$\alpha = 45°$
23	Pitch Cone Radius	Multiply the pitch diameter by 0.707	$C = 0.707D$
24	Face Angle	Subtract the addendum angle from 45°	$\delta = 45° - \theta$
25	Cutting Angle *	Subtract the dedendum angle from 45 degrees	$\zeta = 45° - \varphi$
26	Angular Addendum	Multiply the addendum by 0.707	$K = 0.707\,S$
27	Number of Teeth in Equivalent Spur Gear	Multiply the number of teeth by 1.41	$N' = 1.41N$

*For gears whose teeth are to be milled, see recommendation of the Brown & Sharpe Mfg. Co., Page 41.

O = outside diameter (edge diameter for internal gears),
J = vertex distance,
j = vertex distance at small end,
N' = number of teeth in equivalent spur gear.

Sub$_p$ refers to dimensions applying to pinion (a_p, N_p, etc.)

Sub$_g$ refers to dimensions applying to gear (a_g, N_g, etc.)

It will be noted that directions for the use of these rules are given for each of the six cases of right angle bevel gearing, miter bevel gearing, acute angle and obtuse angle bevel gearing, and crown and

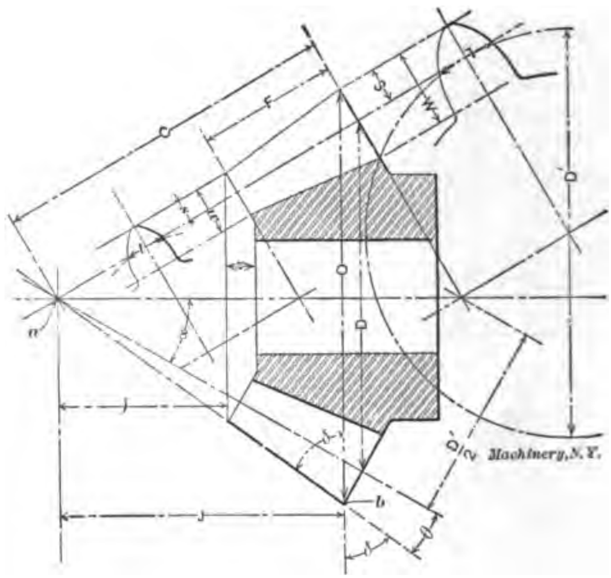


Fig. 12. Diagram Explaining Certain Calculations Relating to Bevel Gears

internal bevel gears. Further instruction as to their use can be obtained from the examples given in Chapter II.

Rules and Formulas for Bevel Gear Calculations

The derivation of most of these formulas is evident on inspection of Figs. 1 to 12 inclusive, for anyone who has a knowledge of elementary trigonometry. It is not necessary to know how they were derived to use them, however, as all that is needed is the ability to read a table of sines and tangents.

Formulas 5, 6, 7 and 8 are the same as for Brown & Sharpe standard gears. The dimensions at the small end of the tooth given by Formulas 10, 11 and 19 obviously are to the corresponding dimensions at the large end, as the distance from the small end of the tooth to the vertex of the pitch cone is to the pitch cone radius. This relation is expressed by these formulas. The derivation of Formula 20 may be understood by reference to Fig. 12:

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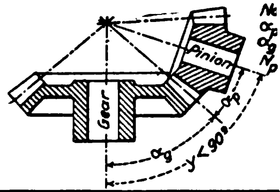
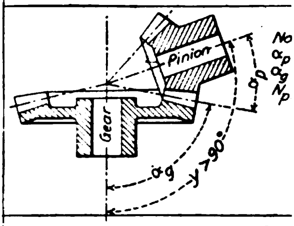
Bevel Gears w



Use Rules and

No.	To Find	
28	Pitch Cone Angle (or Edge Angle) of Pinion	Divide the cosi number teeth in
29	Pitch Cone Angle (or Edge Angle) of Gear	Divide the cosi the num number
30	Proof of Calculations for Pitch Cone Angles	The sur and gea

Bevel Gears



Use Rule

No.	To Find	
31	Pitch Cone Angle (or Edge Angle) of Pinion	Divide th angle by number of teeth minus t
32	Whether Gear is a Regular Bevel Gear, a Crown Gear, or an Internal Bevel Gear	Add 90 a If the su rules ar If the su and for If the su rules a
33	Pitch Cone Angle (or Edge Angle) of Gear	Divide th by the di of teeth in the g the cen

$$D' = \frac{D}{\cos\alpha} = \frac{N}{P \times \cos\alpha}, \text{ also } D' = \frac{N'}{P}$$

$$\text{therefore } \frac{N'}{P} = \frac{N}{P \times \cos\alpha}, \text{ or } N' = \frac{N}{\cos\alpha}.$$

Formula 21 for checking the calculations will also be understood from Fig. 12, where it will be seen that

$$O = 2\,a\,b \times \cos\delta, \text{ also that } a\,b = \frac{C}{\cos\theta},$$

$$\text{therefore } O = \frac{2\,C \times \cos\delta}{\cos\theta}.$$

Formulas 22 to 27 inclusive are simply the corresponding Formulas 1, 9, 14, 15, 16 and 20 when $\alpha = 45$ degrees.

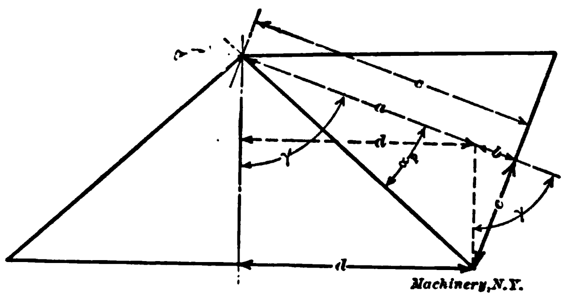


Fig. 13. Diagram for Obtaining Pitch Cone Angle of Acute Angle Gearing

Formula 28 is derived as shown in Fig. 13.

$$c = \frac{e}{\tan\alpha_p}, \text{ also, } c = a + b = \frac{d}{\sin\gamma} + \frac{e}{\tan\gamma},$$

$$\text{therefore, } \frac{e}{\tan\alpha_p} = \frac{d}{\sin\gamma} + \frac{e}{\tan\gamma}.$$

Solving for $\tan\alpha_p$, we have: $\tan\alpha_p = \dfrac{e\,(\sin\gamma \times \tan\gamma)}{d\tan\gamma + e\sin\gamma}$.

Dividing both numerator and denominator by $e\tan\gamma$, we have:

$$\tan\alpha_p = \frac{\sin\gamma}{\dfrac{d}{e} + \dfrac{\sin\gamma}{\tan\gamma}}$$

Since $d = \dfrac{N_g}{2P}$ and $e = \dfrac{N_p}{2P}$, and since $\dfrac{\sin}{\tan} = \cos$, we have:

$$\tan\alpha = \frac{\sin\gamma}{\dfrac{N_g}{N_p} + \cos\gamma}$$

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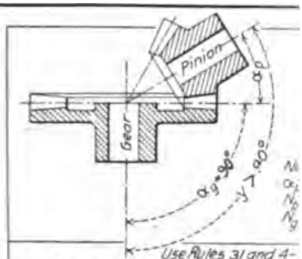
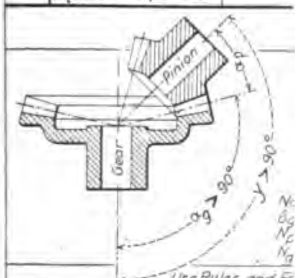


Use Rules 31 and 4-
37, 15 and 38 in the order given t
to find center angle and dimension

No.	To Find	
34	Pitch Cone Angle (or Edge Angle) of Pinion	Divide numbe
35	Center Angle	Add 90
36	Pitch Cone Radius	Divide
37	Face Angle of Gear	The face
38	Number of Teeth in Equivalent Spur Gear	The tee



Use Rules and Fo
Formulas 39, 30, 40, 41, 15, 42, 43,

No.	To Find.	
39	Pitch Cone Angle (or Edge Angle) of Gear	Divide angle, by degrees the num number whose ta
40	Pitch Cone Radius	Divide t degrees
41	Face Angle of Gear	Subtrac cone an
42	Angular Addendum of Gear	Multiply grees mi
43	Outside (or Edge) Diameter of Gear	Subtrac pitch di
44	Number of Teeth in Equivalent Internal Spur Gear	Divide th grees m

Formula 29 is derived by the same process for the other gear. Formula 31 (and likewise 33) is derived from Fig. 14, using the following fundamental equation:

$$\frac{e}{\tan a_p} = \frac{d}{\sin(180^\circ - \gamma)} - \frac{e}{\tan(180^\circ - \gamma)}$$

When solved for $\tan a_p$, this gives Formula 31.

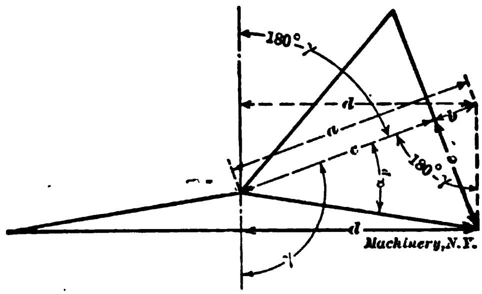


Fig. 14. Diagram for Obtaining Pitch Cone Angle of Obtuse Angle Gearing

Rule 32, of course, simply expresses the operation of finding out whether the pitch cone angle of the gear is less, equal to or greater than 90 degrees. The derivation of Formula 34 is shown in Fig. 15:

$$\sin a_p = \frac{e}{d} = \frac{N_p}{N_g}$$

Since in a crown gear the dimension $\frac{D'}{2}$ in Fig. 12 is to be measured

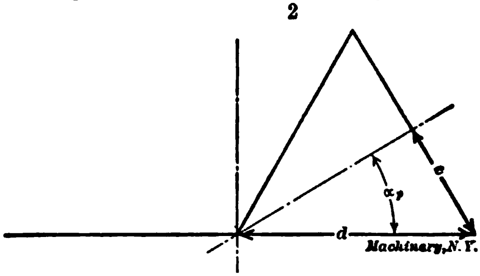


Fig. 15. Diagram for Obtaining Pitch Cone Angle of Pinion to Mesh with Crown Gear

parallel to the axis, and will therefore be of infinite length, the form of the teeth will correspond to those of a spur gear having a radius of infinite length, that is to say, to a rack. This accounts for Formula 38.

Formulas 39, 40, 42 and 44 are simply the corresponding Formulas 33, 9, 16 and 20 changed to avoid the use of negative cosines, etc., which occur with angles greater than 90 degrees. These negative functions might possibly confuse readers whose knowledge of trigonometry is elementary. The other formulas for internal gears are readily comprehensible from an inspection of Fig. 11.

CHAPTER II

EXAMPLES OF BEVEL GEAR CALCULATIONS

A number of examples of calculations are here given for practice, covering all the various types shown in Figs. 3 to 8 inclusive. The conditions of the various examples differ from each other only in the center angle. While such great accuracy is not required in the work itself, it will be found convenient in the calculations to use tables of sines and tangents which give readings for minutes to five figures. This permits accurate checking of the various dimensions by Rules and Formulas 3, 21, etc.

Shafts at Right Angles

Let it be required to make the necessary calculations for a pair of bevel gears in which the shafts are at right angles; diametral pitch = 3, number of teeth in gear = 60, number of teeth in pinion = 15, and width of face = 4 inches.

$$\tan a_p = 15 \div 60 = 0.25000 = \tan 14° \; 2' \qquad (1)$$

$$\tan a_g = 60 \div 15 = 4.00000 = \tan 75° \; 58' \qquad (2)$$

$$\gamma = 14° \; 2' + 75° \; 58' = 90° \qquad (3)$$

$$D_p = 15 \div 3 = 5.000'' \qquad (4)$$

$$S = 1 \div 3 = 0.3333'' \qquad (5)$$

$$S + A = \frac{1.157}{3} = 0.3856'' \qquad (6)$$

$$W = \frac{2.157}{3} = 0.7190'' \qquad (7)$$

$$T = \frac{1.571}{3} = 0.5236'' \qquad (8)$$

$$C = \frac{5}{2 \times 0.24249} = 10.3097'' \qquad (9)$$

$$s = 0.3333 \times \frac{6.31}{10.31} = 0.2040'' \qquad (10)$$

$$t = 0.5236 \times \frac{6.31}{10.31} = 0.3204'' \qquad (11)$$

$$\tan \theta = \frac{0.3333}{10.3097} = 0.03233 = \tan 1° \; 51' \qquad (12)$$

$$\tan \phi = \frac{0.3856}{10.3097} = 0.03740 = \tan 2° \; 9' \qquad (13)$$

$$\delta = 90° - (14° \; 2' + 1° \; 51') = 74° \; 7' \qquad (14)$$

$$\zeta = 14° \; 2' - 2° \; 9' = 11° \; 53' \qquad (15)$$

$$K = 0.3333 \times 0.97015 = 0.3234'' \quad \ldots\ldots (16)$$

$$O = 5.000 + 2 \times 0.3234 = 5.6468'' \quad \ldots\ldots (17)$$

$$J = \frac{5.6468}{2} \times 3.51441 = 9.9225'' \quad \ldots\ldots (18)$$

$$j = 9.9225 \times \frac{6.31}{10.31} = 6.0726'' \quad \ldots\ldots (19)$$

$$N' = \frac{15}{0.97015} = 15.4 \quad \ldots\ldots (20)$$

$$5.6468'' \cong \frac{20.6194 \times 0.27368}{0.99948} = 5.6461'' \quad \ldots\ldots (21)$$

This gives all the data required for the pinion. Rules 5 to 13 inclusive apply equally to the gear and the pinion, so we have only calculations by Rules and Formulas 4 and 14 to 21 to make, though it is well to calculate Formula 9 a second time as a check for the same calculation for the pinion.

$$D = \frac{60}{3} = 20.000'' \quad \ldots\ldots (4)$$

$$C = \frac{20}{2 \times 0.97015} = 10.3077'' \quad \ldots\ldots (9)$$

$$\delta = 90 - (75^\circ\ 58' + 1^\circ\ 51') = 12^\circ\ 11' \quad \ldots\ldots (14)$$

$$\zeta = 75^\circ\ 58' - 2^\circ\ 9' = 73^\circ\ 49' \quad \ldots\ldots (15)$$

$$K = 0.3333 \times 0.24249 = 0.0808'' \quad \ldots\ldots (16)$$

$$O = 20 + 2 \times 0.0808 = 20.1616'' \quad \ldots\ldots (17)$$

$$J = \frac{20.1616}{2} \times 0.2159 = 2.1764'' \quad \ldots\ldots (18)$$

$$j = 2.1764 \times \frac{6.31}{10.31} = 1.3320'' \quad \ldots\ldots (19)$$

$$N' = \frac{60}{0.24249} = 247 \quad \ldots\ldots (20)$$

$$20.1616'' \cong \frac{20.6154 \times 0.97748}{0.99948} = 20.1615'' \quad \ldots\ldots (21)$$

This gives the calculations necessary for this pair of gears, which are shown drawn and dimensioned in Fig. 19. There are two or three other dimensions, such as the over-all length of the pinion, etc., which depend on arbitrary dimensions given the gear blank. Directions for calculating these are given in the text in connection with Fig. 19.

Acute Angle Bevel Gearing

Let it next be required to calculate the dimensions of a pair of bevel gears whose center angle is 75 degrees, the number of teeth in the pinion 15, the number of teeth in the gear 60, the diametral pitch 3, and the width of face 4 inches. This is the same as the first example,

except for the center angle. Fo
chart we have:

$$\tan a_p = \frac{0.96593}{\frac{60}{15} + 0.25882} = 0.226$$

$$\tan a_g = \frac{0.96593}{\frac{15}{60} + 0.25882} = 1.898$$

$$\gamma = 12^\circ\ 47' + 62^\circ\ 13' =$$

Formulas 4 to 8 as in first exa
$t = 0.3382''$, $\theta = 1^\circ\ 41'$, $\phi = 1^\circ$
$0.3251''$, $O = 5.6502''$, $J = 10.9501''$,

$$5.6502'' \cong \frac{22.598 \times 0.24982}{0.99957} = 5$$

For the gear, the additional calc
$\zeta = 60^\circ\ 16'$, $K = 0.1553''$, $O = 20.3$
129.

$$20.3106'' \cong \frac{22.606 \times 0.89803}{0.99957} = 2$$

The above calculations are not
are merely re-duplications of forn

Crow

Suppose it is required to make
same number of teeth, pitch and fa
are the additional calculations ne
mulas in the order given by the c

$$\sin a_p = \frac{15}{60} = 0.25000 = \sin 14$$

$$\gamma = 90^\circ + 14^\circ\ 29' = 104^\circ$$

The other calculations are simil

Internal

Let it be required to design a p
ber of teeth, pitch and face, in wl
This being an example of obtuse a

$$\tan a_p = \frac{0.90631}{\frac{60}{15} - 0.42262} = 0.2533$$

Thus, according to Rule 32, we

$14^\circ\ 13' + 90^\circ = 104^\circ\ 13'$ 11

showing that the gear is an internal bevel gear. Applying the rules and formulas for internal bevel gearing, we have:

$$\tan a_s = \frac{0.90631}{0.42262 - \frac{15}{60}} = 5.25032 = \tan 79^\circ\ 13'$$

$$180^\circ - 79^\circ\ 13' = 100^\circ\ 47' \ldots\ldots (39)$$

$$\gamma = 100^\circ\ 47' + 14^\circ\ 13' = 115^\circ \ldots\ldots (30)$$

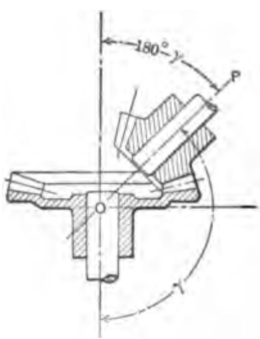
$$C = \frac{20}{2 \times 0.98234} = 10.1797'' \ldots\ldots (40)$$

$$\delta = 100^\circ\ 47' + 1^\circ\ 53' - 90^\circ = 12^\circ\ 40' \ldots\ldots (41)$$

$$\zeta = 98^\circ\ 37', \text{ and } K = 0.0624''$$

$$O = 20 - 2 \times 0.0624 = 19.8752'' \ldots\ldots (43)$$

$$N' = \frac{60}{0.1871} = 320 \text{ (internal)} \ldots\ldots (44)$$



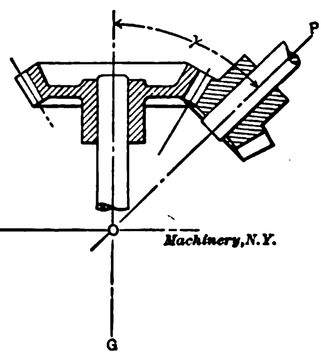


Fig. 16. Internal Bevel Gearing

Fig. 17. Acute Bevel Gearing Used as Substitute for Internal Gearing in Fig. 16

The calculations for the pinion and the other calculations for the gear are similar to those already given.

Obtuse Angle Bevel Gearing

Let it be required to calculate the dimensions of the same set of gears but with the center angle of 100 degrees. This being an example of obtuse angle gearing, we apply Formula 31 as follows:

$$\tan a_p = \frac{0.98481}{\frac{60}{15} - 0.17365} = 0.25738 = \tan 14^\circ\ 26' \ldots\ldots (31)$$

and thus discover that it is an example of regular obtuse angle gearing, since

$$14^\circ\ 26' + 90^\circ = 104^\circ\ 26' > 100^\circ \ldots\ldots (32)$$

The remaining calculations for the angles are as follows:

$$\tan \alpha = \frac{0.98481}{\frac{15}{60} - 0.17365} = 12.898($$

$$\gamma = 14° 26' + 85° 34' = 1$$

and the calculations for the other

How to Avoid Int

When Rule 32, in any given case
an internal bevel gear, such as shov
be avoided without changing the]
of the teeth in the gear, the pitch
This is done simply by subtractin
degrees, and using the remainder a
a set of acute angle gears by Rul(
pair of bevel gears calculated on t
Fig. 16 is shown in Fig. 17. It v
place on the other side of the axis (

It is necessary to avoid internal
possible to cut them. It may be t
machines will do this work, if the
but no form of generating machin
if any one has ever cut a pair o
writer has seen occasional example:

CHAPTER III

SYSTEMS OF TOOTH OUTLINES USED FOR BEVEL GEARING

Five systems of tooth outlines are commonly used for bevel gearing. They are the cycloid, the standard 14½-degree involute, the 20-degree involute and the 15- and 20-degree octoid.

The Cycloidal System

The cycloidal form of tooth is obsolete for cut bevel gears, and is rarely met with nowadays for cast gears even. It requires very careful workmanship, and is difficult or impossible to generate. It is also a bad shape to form with a relieved cutter, as the cutting edge tends to drag at the pitch line, where for a short distance the sides of the teeth are nearly or quite parallel. For spur gearing it has a few points of advantage over the involute form of tooth, but in the case of bevel gearing these are nullified by the impossibility of generating the teeth in practicable machines. The cycloidal form of tooth need not be seriously considered for bevel gears.

Involute and Octoid Teeth

Most bevel gears are made on the involute system, of either the standard 14½-degree pressure angle, or the 20-degree pressure angle. In spur gear teeth the pressure angle may be defined as the angle which the flat surface of the rack tooth makes with the perpendicular to the pitch line. The 20-degree tooth is consequently broader at the base and stronger in form than the 14½-degree tooth. This same difference applies to bevel gears. Most bevel gears that are milled with formed cutters are made to the 14½-degree standard, as cutters for this shape are regularly carried in stock. The planed gears, made by the templet or generating principles, are nowadays often made to the 20-degree pressure angle, both for the sake of obtaining stronger teeth, and for avoiding undercutting of the flanks of the pinions as well. This undercutting is due to the phenomenon of "interference," as it is called, which is minimized by increasing the pressure angle.

If you ask the manufacturer to plane a pair of involute bevel gears for you on the Bilgram, Gleason or other similar generating machine, he will not give you involute teeth, but something "just as good." This "just as good" form was invented by Mr. Bilgram, and was named "Octoid" by Mr. Geo. Grant. In generating machines the teeth of the gears are shaped by a tool which represents the side of the tooth of an imaginary crown gear. The cutting edge of the tool is a straight line, since the imaginary crown gear has teeth whose sides are plane surfaces. It can be shown that the teeth of a true involute crown

gear have sides which are very slightly curved. The minute difference between the tooth shapes produced by a plane crown tooth and a slightly curved crown tooth is the minute difference between the octoid and involute forms. Both give theoretically correct action. The customer in ordering gears never uses the word "octoid," as it is not a commercial term; he calls for "involute" gears.

Formed Cutters for Involute Teeth

For 14½-degree involute teeth, the shapes of the standard cutter series furnished by the makers of formed gear cutters are commonly used. There are 8 cutters in the series, to cover the full range from the 12-tooth pinion to a crown gear. The various cutters are numbered from 1 to 8, as given in the table below:

No. 1 will cut wheels from 135 teeth to a rack.
No. 2 will cut wheels from 55 teeth to 134 teeth.
No. 3 will cut wheels from 35 teeth to 54 teeth.
No. 4 will cut wheels from 26 teeth to 34 teeth.
No. 5 will cut wheels from 21 teeth to 25 teeth.
No. 6 will cut wheels from 17 teeth to 20 teeth.
No. 7 will cut wheels from 14 teeth to 16 teeth.
No. 8 will cut wheels from 12 teeth to 13 teeth.

It should be remembered that the number of teeth in this table refers to the number of teeth in the equivalent spur gear, as given by Rule 20, which should always be used in selecting the cutter used for milling the teeth of bevel gears. Thus for the gear in the first example in Chapter II, the No. 1 cutter should be used. The standard bevel gear cutter is made thinner than the standard spur gear cutter, as it must pass through the narrow tooth space at the inner end of the face. As usually kept in stock, these cutters are thin enough for bevel gears in which the width of face is not more than one-third the pitch cone radius. Where the width of face is greater, special cutters have to be made, and the manufacturer should be informed as to the thickness of the tooth space at the small end; this will enable him to make the cutter of the proper width.

Special Forms of Bevel Gear Teeth

In generating machines (such as the Bilgram and the Gleason) it is often advisable to depart from the standard dimensions of gear teeth as given by Rules and Formulas 1 to 44. For instance, where the pinion is made of bronze and the gear of steel, the teeth of the former can be made wider and those of the latter correspondingly thinner, so as to somewhere nearly equalize the strength of the two. Again, where the pinion has few teeth and the gear many, it may be advisable to make the addendum on the pinion larger and the dedendum correspondingly smaller, reversing this on the gear, making the addendum smaller and the dedendum larger. This is done to avoid interference and consequent undercut on the flanks of pinions having a small number of teeth. Such changes are easily effected on generating machines and instructions for doing this for any case will be furnished by the makers.

CHAPTER IV

STRENGTH AND DURABILITY OF BEVEL GEARS

The same materials are used in general for making bevel gears as for spur gears and each has practically the same advantages and disadvantages for both cases. In general, the strength of different materials is roughly proportional to the durability.

The Materials Used for Making Bevel Gears

Cast iron is used for the largest work, and for smaller work which is not to be subjected to heavy duty. In cases where great working stress or a sudden shock is liable to come on the teeth, steel is ordinarily used. Such gears are made from bar stock for the smallest work, from drop forgings for intermediate sizes made on a manufacturing basis, and from steel castings for heavy work. The softer grades of steel are not fitted for high-speed service, as this material abrades more rapidly than cast iron. This objection does not apply to hardened steels, such as used in automobile transmission gears.

As in the case of spur gearing it is quite common to make the gear and pinion of different materials. This is advantageous from the standpoint of both efficiency and durability, since two dissimilar metals work on each other with less friction than similar metals, as is well known. Cast iron and steel, and steel and bronze are common combinations. In general, the pinion should be made of the stronger material, since it is of weak form; and it should be made of the more durable material, as it revolves more rapidly and each tooth comes into working contact more times per minute than do those of the larger mating gear. In a steel and cast iron combination, then, the pinion should be of steel, while the gear is of cast iron. In a steel and bronze combination, the pinion should be of steel and the gear of bronze, though this is more costly than when the materials are reversed.

A wide range of physical qualities is now available in steel, both for parts small enough to be made from bar stock, and for those made from drop forgings. Recent improvements have also given almost as much flexibility in the choice of steel castings. Gears made from high grade steels may be subjected to heat treatments which increase their durability and strength amazingly.

Raw-hide and fiber are quite largely used for pinion blanks in cases where it is desired to run gearing at a very high speed and with as little noise as possible. There is a little more difficulty in building up a raw-hide blank properly for a bevel gear than for a spur gear. Fiber, which is used in somewhat the same way, has the merit of convenience and comparative inexpensiveness, as it may be purchased in a variety of sizes of bars, rods, tubes, etc., ready to be worked up into pinion blanks at short notice. It is not so strong as raw-hide,

STRENGTH OF

List of Refe.

D - pitch diameter of gear in inches.

R - revolutions per minute.

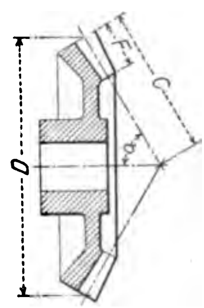
V - velocity in ft. per min. at pitch diameter.

S_s - allowable static unit stress for material.

S - allowable unit stress for material at given velocity.

F - width of face.

N' - No. of teeth in equivalent spur gear (See diagram).



$N' = \frac{\text{Number of teeth}}{\text{Cos } \alpha}$

(Rule No. 20)

Use rules and formul

No.	To Find	
45	Velocity in ft. per min. at the pitch diameter	Multiply the product o the number of revoluti
46	Allowable unit stress at given velocity	Multiply the allowa and divide the res feet per minute pl
47	Maximum safe tangential load at pitch diameter	Multiply together the o velocity, the width of fa and the difference betw the width of face; divid the diametral pitch o
48	Maximum safe Horse Power	Multiply the safe load at in feet per minute, anc

and is difficult to machine owing to its gritty composition. For light duty at high speed it does very well. For large, high-speed gearing it was formerly a common practice to use inserted wooden teeth on the gear, meshing with a solid cast iron pinion. This construction is seldom used for cut gearing.

Strength of Bevel Gear Teeth

The Lewis formula is the one generally used in this country for calculating the strength of gears. Mr. Myers, who had an article on the "Strength of Gears" in the December, 1906, issue of MACHINERY, gives Mr. Barth's adaptation of this formula for calculating the strength of bevel gears. The rules and formulas on the next page are condensed from the method given in the article referred to.

The factors to be taken into account are the pitch diameter of the gear, the number of revolutions per minute, the diametral pitch (or circular pitch as the case may be) the width of face, the pitch cone radius, the number of teeth in the gear and the maximum allowable static fiber stress for the material used. From this we may find the maximum allowable load at the pitch line, and the maximum horse-power the gear should be allowed to transmit.

The reader familiar with the Lewis formula will note that Rule and Formula 47 is the same as for spur gears with the exception of the additional factor $\frac{C - F}{C}$. This factor is an approximate one which expresses the ratio of the strength of a bevel gear to that of a spur gear of the same pitch and number of teeth, the decrease being due to the fact that the pitch grows finer toward the vertex. This factor is approximate only and should not be used for cases in which F is more than $1/3\,C$; but since no bevel gears should be made in which F is more than $1/3\,C$, the rule is of universal application for good practice. As the width of face is made greater in proportion to the pitch cone radius, the increase of strength obtained thereby grows proportionately smaller and smaller, as may be easily proved by analysis and calculation. Actually the advantage of increasing the width of face is even less than is indicated by calculation, since the unavoidable deflection and disalignment of the shaft is sure at one time or another to throw practically the whole load on the weak inner ends of the teeth, which thus have to carry the load without help from the large pitch at the outer ends.

Rules and Formulas for the Strength of Bevel Gears

The reference letters, rules and formulas on the next page, for the strength of bevel gear teeth, are self-explanatory. As an approximate guide for ordinary calculations, 8,000 pounds per square inch may be allowed for the static stress of cast iron and 20,000 pounds for ordinary machine steel or steel castings. Where the gearing is to be subjected to shock, 6,000 pounds for cast iron and 15,000 pounds for steel are more satisfactory figures. The wide range of materials offered the designer, however, makes any fixed tabulation of fiber stress impracti-

cable. An example showing the use
given herewith.

Calculate the maximum load at th
allowed for the bevel gears in Fig. 19
stress for the pinion is 20,000 pound
per square inch; the pinion runs at
calculations for the pinion are as foll

$$V = 0.262 \times 5 \times 300 = 400 \text{ feet p}$$

$$S = 20,000 \times \frac{600}{600 + 400} = 12,00($$

$$W = \frac{12,000 \times 4 \times 0.289 \times 6.3}{3 \times 10.3} =$$

For the gear, the velocity is the sa
sary calculations are as follows:

$$S = 8,000 \times \frac{600}{600 + 400} = 4,800 \text{ p}$$

$$W = \frac{4,800 \times 4 \times 0.421 \times 6.3}{3 \times 10.3} = 1,($$

The gear is, therefore, the weaker
allowable tooth pressure. The maxim
transmit safely is found as follows:

$$\text{H. P.} = \frac{1,650 \times 400}{33,000} = 20 \;\ldots\ldots$$

Durability is practically of as mu(
portioning bevel gears, but unfortun
for making satisfactory comparisons
cedure is followed of designing the
to providence that they will not we
machine in which they are used.

CHAPTER V

DESIGN OF BEVEL GEARING

So far we have dealt with design as relating to calculations. In this chapter will be discussed the application of the calculated dimensions, the determination of the factors left to the judgment, and the recording of the design in the drawing.

Bevel Gear Blanks

Various forms may be given to the blanks or wheels on which bevel gear teeth are cut, depending on the size, material, service, etc., to be provided for. The pinion type of blank is shown in Fig. 12 and elsewhere. It is used mostly, as indicated by the name, for gears of a small number of teeth and small pitch cone angle. Where the diameter of the bore comes too near to the bottoms of the teeth at the

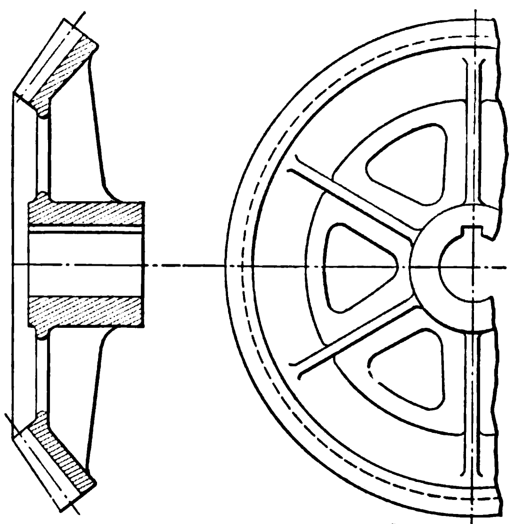


Fig. 18. T-arm Style of Bevel Wheel for Heavy Work

small end, it is customary to omit the recess indicated by dimension z, and leave the front face of the pinion blank as in the case shown in Fig. 19.

For gears of a larger number of teeth, the web type shown in Fig. 9 and elsewhere is appropriate. This does not require to be finished all over, as the sides of the web, the outside diameter of the hub, and the under side of the rim may be left rough if desired.

A steel gear suitable for very heavy work is shown in Fig. 18. Here the web is reinforced by ribs. The web may be cut out so that the

rim is supported by T-shaped arms, as shown. This makes a very stiff wheel and at the same time a very light one, when its strength is considered. Where the pitch cone angle is so great that the strengthening rib would be rather narrow at the flange, it may be given the form shown in Fig. 19 in place of that shown in Fig. 18.

General Considerations Relating to Design

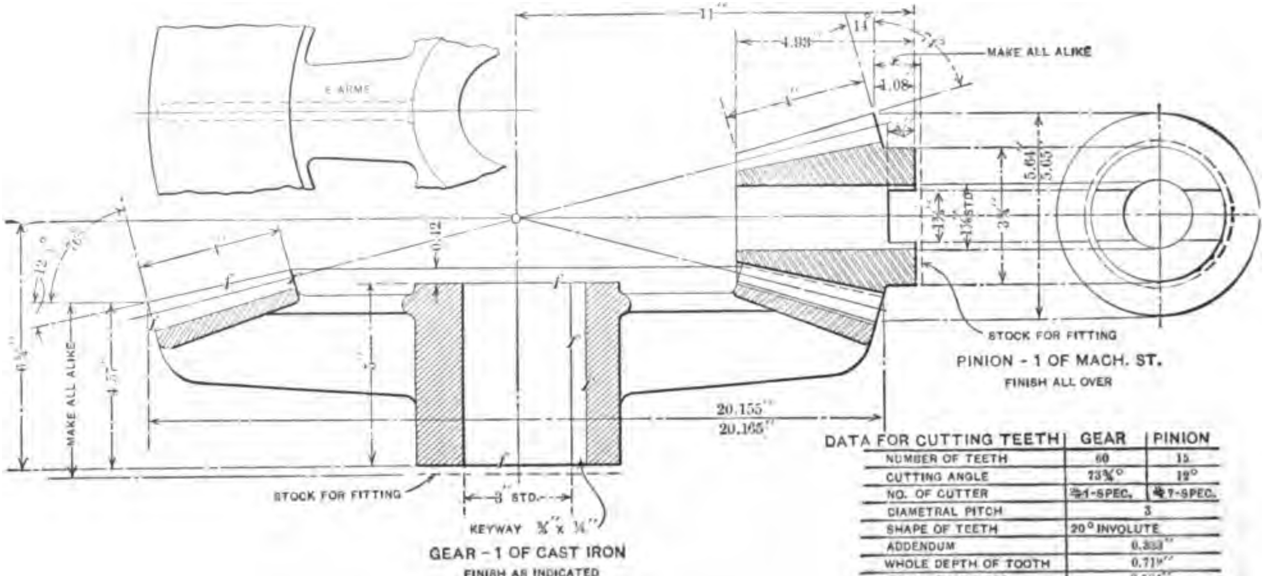
The performance of the most carefully designed and made bevel gears depends to a considerable extent on the design of the machine in which they are used. When the shafts on which a pair of bevel gears are mounted are poorly supported or poorly fitted in their bearings, the pressure of the driving gear on the driven, causes it to climb up on the latter, throwing the shafts out of alignment. This in turn causes the teeth to bear with a greater pressure at one end of the face (usually on the outer end) than the other, thus making the tooth more liable to break than is the case where the pressure is more evenly distributed. It is important, therefore, to provide rigid shafts and bearings and careful workmanship for bevel gearing.

The question of alignment of the shafts should be considered in deciding on the width of face of the gear. Making the width of the face more than one-third of the pitch cone radius adds practically nothing to the strength of the gear even theoretically, since the added portion is progressively weaker as the tooth is lengthened, as has been explained. In addition to this, there is the danger that through springing of the shafts or poor workmanship, the load will be thrown onto the weak end of the tooth, thus fracturing it. For this reason it may be laid down as a definite rule that there is nothing to be gained by making the face of the bevel gear more than one-third of the pitch cone radius, as required by Rules 45 to 48.

The Brown & Sharpe gear book gives a rule for the maximum width of face allowable for a given pitch. The width of face should not exceed five times the circular pitch or 16 divided by the diametral pitch. This rule is also rational since the danger to the teeth from the misalignment of the shaft increases both with the width of face and with the decrease of the size of the tooth, so that both of these should be reckoned with. In designing gearing it is well to check the width of face from the rule relating to the pitch cone radius and that relating to the pitch as well, to see that it does not exceed the maximum allowed by either.

Model Bevel Gear Drawing

It is not enough for the designer to carefully calculate the dimensions of a set of bevel gearing. In addition to this he has the important task of recording these dimensions in such a form that they will be intelligible to an intelligent workman, and will plainly furnish him every point of information needed for the successful completion of the work without further calculation. A drawing which practically fills these requirements is shown in Fig. 19. The arrangement of this drawing and the amount and kind of information shown on it are based on the drafting-room practice of the Brown & Sharpe



DATA FOR CUTTING TEETH	GEAR	PINION
NUMBER OF TEETH	60	15
CUTTING ANGLE	73¾°	12°
NO. OF CUTTER	1-SPEC.	7-SPEC.
DIAMETRAL PITCH	3	
SHAPE OF TEETH	20° INVOLUTE	
ADDENDUM	0.333"	
WHOLE DEPTH OF TOOTH	0.719"	
TOOTH THICKNESS	0.524"	
ADDENDUM AT SMALL END	0.204"	
TOOTH THICKNESS " "	0.320"	

Machinery, N.Y.

Fig. 19. Model Working Drawing for Bevel Gears to be Cut with Formed Milling Cutter

Mfg. Co., as described by Mr. Burlingame in the article "Figuring Gear Drawings" in the August, 1906, issue of MACHINERY. Some changes and additions have been made in the arrangement of the dimensioning, however, so that firm cannot be held responsible for all that appears on the engraving.

In general, the dimensions necessary for turning the blank have been given on the drawing itself, while those for cutting the teeth are given in tabular form. All the dimensions were calculated from Rules 1 to 21 inclusive and may be checked for practice by the reader. It will be noticed that limits are given for the important dimensions. This should always be done for manufacturing work which is inspected in its course through the shop. It ought to be done even when a single gear is made, as it is exceedingly difficult to properly set a gear if the workman does not work close enough. There is no sense, however, in asking him to work to thousandths of an inch on blanks like these, so he should be given some notion as to the accuracy required by limits such as shown.

It is assumed that the gears are to be cut with rotary cutters. It is unusual to do this with a pitch as coarse as this, though there are machines on the market capable of handling such work. In gear cutting machines using form cutters, the blanks are located for axial position by the rear face of the hub. It is necessary also to leave stock at this place for fitting the gears in the machine. It will be seen that the dimension for bevel gears and pinions from the outside edge of the blank to the rear face of the hub is marked "Make all alike." This means that the same amount of stock should be left on all the gears in a given lot so that after the machine is set for one of them, it will not be necessary to alter the adjustment for the remainder.

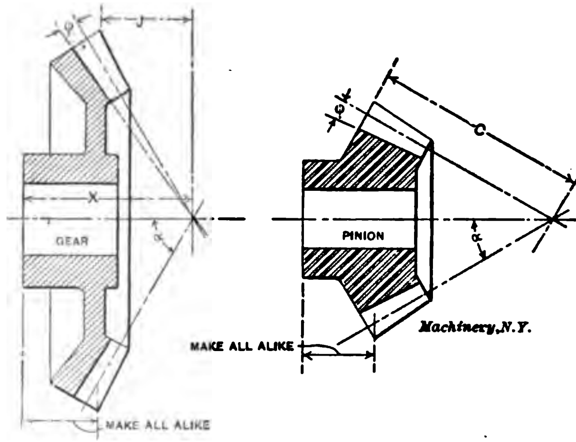
There are one or two dimensions which are not given directly by Rules 1 to 21. One of these is the distance 4.57 inches from the outside edge of the teeth to the finished rear face of the hub of the gear. This dimension is commonly scaled from an accurate drawing, but it may be calculated by subtracting the vertex distance from the distance between the pitch cone vertex O and the rear face of the hub. This gives $6\frac{3}{4} - 2.1764$ equals 4.57 inches (about) as dimensioned. Another dimension not directly calculated is the over-all length of the pinion. This may be obtained by subtracting the vertex distance at the small end (j) from the distance between the vertex and the rear face of the hub, giving 4.98 inches as shown.

In the tabular dimensions for cutting the teeth, most of the figures are self-explanatory. The fact that in this particular case a 20-degree form of tooth has been adapted to avoid the undercut in small pinions (see Chapter III on Systems of Tooth Outlines used for Bevel Gearing) is indicated in the table.

The number of cutter is selected from the table on page 21 in accordance with the number of teeth (N') in equivalent spur gear, as determined by Rule 20. This is 15.4 for the pinion, and 247 for the gear, giving a No. 1 and No. 7 cutter respectively. These cutters are marked

special, owing to the fact that they are 20 degrees involute instead of 14½ degrees. They would be special under any circumstances, however, since the width of face for these gears (4 inches) is more than 1/3 the pitch cone radius, which figures out to 10.3097 inches. Standard bevel gear cutters are only made thin enough to pass through the teeth at the small end when the width of face is not more than 1/3 the pitch cone radius. For this reason cutters thinner than the standard would have to be used.

In bevel pinions of the usual form, such as shown in Fig. 12, dimension *z* there given has to be furnished. This may be scaled from a carefully made drawing, or may be calculated by subtracting the length of the bore of the pinion from the over-all length, the latter being obtained as described for the pinion in Fig. 19. Such dimensions do



Figs. 20 and 21. Additional Dimensions for Gears to be Cut by the Templet Planing Process, or on the Gleason Generating Machine

not need to be given in thousandths on moderately large work. It is also not necessary to give the angles any closer than the quarter degree, as few machines are furnished with graduations which can be read finer than this. In order to check the calculations carefully, however, it is wise, as previously described, to make them with considerable accuracy, using tables of sines and tangents which read to five figures. After the dimensions are calculated, they may be put in more approximate form for the drawing.

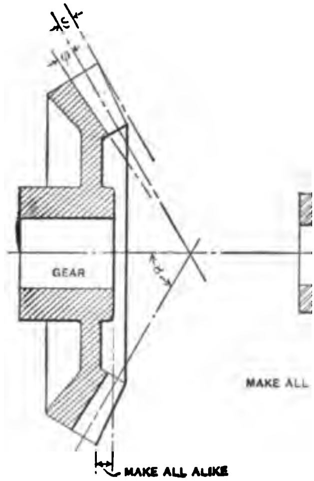
The gear drawing in Fig. 19 is dimensioned more fully, perhaps, than is customary, especially in shops having a large gear-cutting department, where the foreman and operators are experienced and have access to tables and records of data for bevel gear cutting. Every dimension given is useful, however, and it is a good plan to include them all, especially on large work.

Dimensioning Drawings for Gears whose Teeth are to be Planed

The machine on which the teeth of a gear are to be cut determines to some extent the dimensions which the workman needs, so this

should be taken into account in m
which are to be cut on a templet
given in Fig. 19 may be followed
are needed, however, to set the blan
cone corresponds with the central a
with pitch cone angle greater than
from dimension X, as given in Fig.
For gears smaller than 45 degrees, C

There are two commercial forms
general use in this country for pl
These are the Gleason and Bilgram
supporting the gears are different, th
to suit, if it is known beforehand h



Figs. 22 and 23. Additional Dimension
Generating

Gleason machine the dimensioning
be given, in addition to that shown
the pitch cone angle and dedendum
put in the table of dimensions inst
tance from the outside corner of t
hub should be made alike for all s
as for gears which are to be cut by
process. The cutting angle may be

The method of dimensioning for t
in Figs. 22 and 23. Angles a and
before. Dimension S is used for se
angle, and dimension C or the pitc
pitch cone angle (less than 45 deg
both of these dimensions for both ge
may be checked by two different met
sion to be marked "Make all alike" s

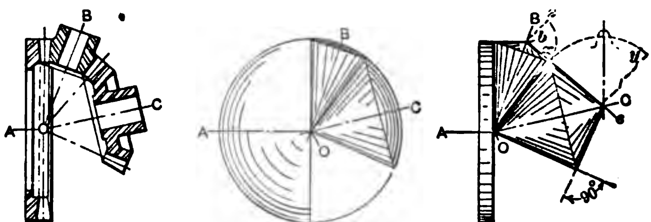
CHAPTER VI

MACHINES FOR CUTTING BEVEL GEAR TEETH

While a very large number of machines have been placed on the market, first and last, for cutting the teeth of bevel gears, the number of designs in common use in this country is small, it being possible, practically, to number them with the fingers of one hand. A brief discussion will here be given of the principles and mechanism of the more commonly used of these machines.

Spherical Basis of the Bevel Gear; Tredgold's Approximation

The principles in common use for cutting teeth of bevel gears are identical with those for cutting the teeth of spur gears, but they are modified in their application to correspond with the spherical basis of the bevel gear. Fig. 24 shows two bevel gears and a crown gear with axes *OC*, *OB*, and *OA* respectively. Fig. 25 shows their pitch surfaces, all of which converge at vertex *O*. These pitch surfaces are formed



Machinery, N.Y.

Fig. 24 **Fig. 25** **Fig. 26**

Illustrating the Spherical Basis of Bevel Gears, and Tredgold's Approximation for Developing the Outlines of the Teeth on a Plane Surface

of cones, cut from a sphere as shown, whose center is at the vertex *O*. The pitch surface of the crown gear becomes the plane face of the hemisphere at the left of Fig. 25. To study the action of these gears the same way as we do that of spur gears when their teeth are drawn on the plane surface of the drawing board, the corresponding lines for the bevel gears would have to be drawn on the surface of the sphere from which the pitch cones were cut. The various pitch circles would be struck from centers located at the points where the axes *OA*, *OB*, and *OC* break with the surface of the sphere. The method of drawing would be identical with that for spur gears. It should be noted that straight lines, on spherical surfaces, are represented by great circles—that is to say, by the intersection of the surface with planes passing through the center of the sphere.

Owing to the impracticability of the sphere as a drawing board, a process, known as "Tredgold's Approximation," is usually followed for laying out the teeth of bevel gears. This is shown in Fig. 26 applied to the same case as in the two preceding figures. The teeth are drawn

and the action studied on surfaces
cones—that is, on the cones with
of these cones can be developed oı
axes *OB* and *OC*. In these cases t
there illustrated. Teeth drawn oı
may be laid out on the conical surf
gear teeth. Teeth so drawn are id
spur gear illustrated in Fig. 12, :
with Fig. 26. For the crown gear,
surface of the cylinder.

Principles of Action of Be

There are three principles of ac
teeth of bevel gears, namely, the f
ing-generating principles. There :

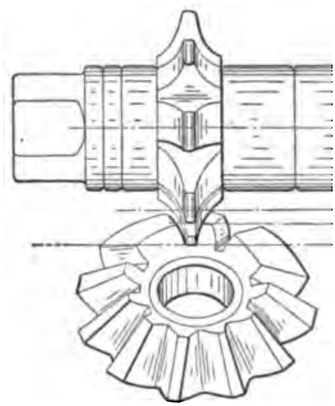


Fig. 27. Shaping the Teeth of a Bev

tent in Europe which employ a fou
principle. It is not in use in this
here.

The *formed tool principle* is il
cutter is shown shaping one side o
blank is tipped up to cutting an
the direction of the arrow. It wil
nation of the figure that the form
mate. It is evident that the rig
ducing its own unchanging outlin
of the tool at the right. This fo
has been explained, the teeth and
towards the apex of the pitch c
is evident that the outline of a
same as that at the large end, t
of the exact outline at the large

process and as shown in the figure. The method of adjusting the cuts to approximate the desired shape is described in the next chapter.

The Templet Principle: This principle is illustrated in Fig. 28, in skeleton form only. A former or templet is used which has the same outline as would a tooth of the gear being cut, if the latter were extended as far from the apex of the pitch cone as the position in which the former is placed. The tool is carried by a slide which reciprocates it back and forth along the length of the tooth in a line of direction (OX, OY, etc.) which passes through vertex O of the pitch cone. This slide may be swiveled in any direction and in any plane about this vertex, and its outer end is supported by the roller on the former. With this arrangement, as the slide is swiveled inward about the vertex, the roll runs up on the templet, raising the slide and the tool so as to reproduce on the proper scale the outline of the former on the tooth being cut. Since the movement of the tool is always toward

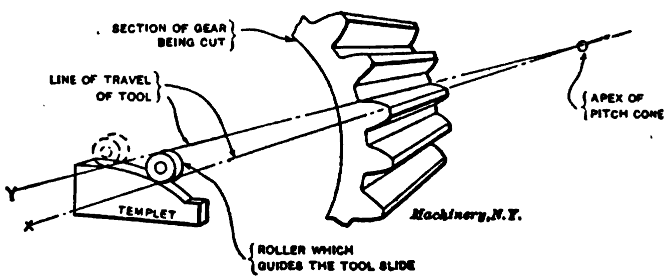


Fig. 28. Illustrating the Templet Principle for Forming the Teeth of Bevel Gears

the vertex of the pitch cone, the elements of the tooth vanish at this point and the outlines are similar at all sections of the tooth, though with a gradually decreasing scale as the vertex is approached—all as required for correct bevel gearing.

The arrangement thus shown diagrammatically is modified in various ways in different machines, but the movement imparted to the tool in relation to the work is the same in all cases where the templet principle is employed, no matter what the connection between the templet and the tool may be.

The Mold-generating Principle: Suppose we have a bevel gear blank made of some plastic material, such as clay or putty. By transposing Formula 34, $\sin \alpha_p = \dfrac{N_p}{N_g}$ to read $N_g = \dfrac{N_p}{\alpha_p}$, it is evidently possible to make a crown gear which will mesh properly with any bevel gear, such as the one we wish to form. If this crown gear and the plastic blank are properly mounted with relation to each other and rolled together, the tooth of the crown gear will form tooth spaces and teeth of the proper shape in the blank. This is the foundation principle of the molding-generating method.

In practice we have blanks of solid steel or iron to machine instead of putty or clay, so the operation has to be modified accordingly. Fig.

29 shows in diagrammatic form an apparatus for using the shaping or planing operation with the molding-generating principle. Here the crown gear is of larger diameter than is required to mesh with the gear being cut, and it engages a master gear keyed to the same shaft as the gear being cut, and formed on the same pitch cone. If the teeth of the crown gear, instead of being comparatively narrow as shown, were extended clear to the vertex *O*, they would mesh properly with the gear to be cut. The tooth is provided as shown having a line of movement such that the point of the tooth travels in line *OX*, which is the corner of a tooth of an imaginary extension of the crown gear. This crown gear has a plane face (see reference to "octoid" form of tooth on page 20) and the cutting edge of the tooth is straight and

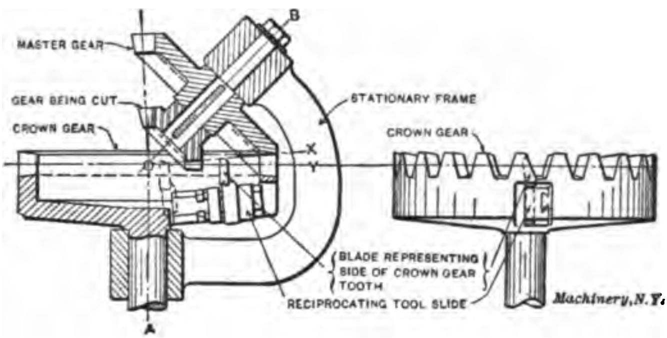


Fig. 29. Model Illustrating the Planing or Shaping Operation Applied to the Molding-generating Principle of Forming Teeth of Bevel Gears

set to mesh the face of the tooth. As it is reciprocated by suitable mechanism (not shown) the cutting edge represents a face of the imaginary crown gear tooth. If now, the master gear and crown gear are rolled together and the tool reciprocating starts in at one side of the gear to be cut and passing out at the other, the straight cutting edge of the tooth will generate one side of a tooth in the gear to be cut in the same way as if the extended tooth of the crown gear were rolling its shape on one side of the tooth of a plastic blank. This simple mechanism has, of course, to be complicated by provisions for cutting both sides of the tooth, and for indexing the work from one tooth to the other so as to complete the entire gear. Arrangements have to be made also to make the machine adjustable for bevel gears of all angles, numbers of teeth and diameters within its range.

The use of the three principles illustrated in Figs. 27, 28 and 29 is not limited to the cutting operation shown for each case. In Fig. 27, for instance, a formed planer or shaper tool may be used as well as a formed milling cutter. Templet machines have been made in which a milling cutter is used instead of a shaper tool. This is true also of the molding-generating principles shown in Fig. 29.

Machines for Cutting the Teeth of Bevel Gears by the Formed Tool Process

A very common method of using the formed tool for cutting bevel gears makes use of the ordinary plain or universal milling machine and adjustable dividing head. Cutting bevel gears by this method is described in the next chapter, so it will not be described here.

Most builders of automatic gear cutting machines furnish them, if desired, in a style which permits the swiveling of the cutter slide or of the work spindle to any angle from 0 to 90 degrees, thus permitting the automatic cutting of bevel gears by the formed cutter process.

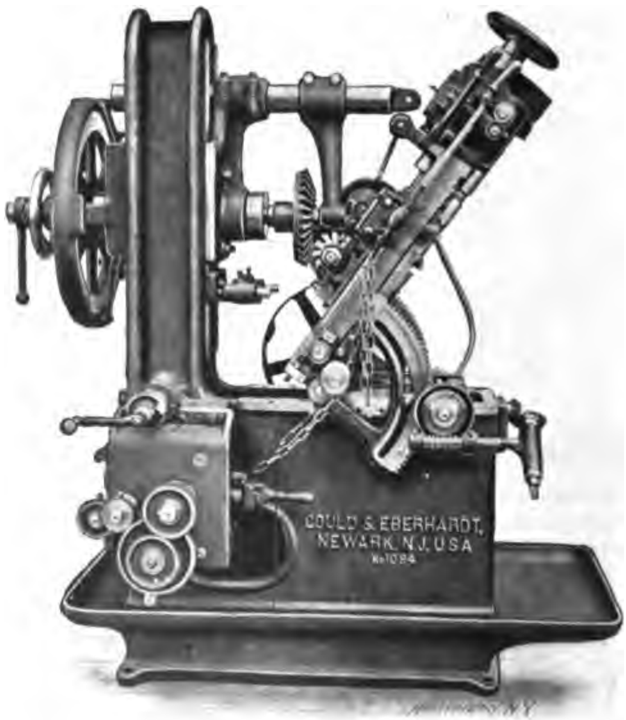


Fig. 30. Gould and Eberhardt Automatic Machine Cutting a Bevel Gear

An example of such a machine is shown in Fig. 30. Here the cutter slide is mounted on an adjustable swinging sector, as may be seen. As explained in the next chapter, it is necessary when cutting bevel gears, to cut first one side of the teeth all around and then the other. Between the two cuts the relation of the work and cutter to each other, as measured in a direction parallel to the axis of the cutter spindle, has to be altered. In the automatic machine this is effected by shifting the cutter spindle axially when the second cut around on the other side of the teeth is taken. Suitable graduations are provided for the angular and longitudinal adjustments.

Bevel Gear Temp

The templet planing machine r is shown in one of the smaller s by a holder reciprocated by an a The slide which carries this tool plane about the horizontal axis which carries the whole mechani ing gearing, etc. This head, in t about a pivot in the bed. The c ment are easily seen in the illust cal and horizontal axes of adjust in front of the tool-slide) is the p

Fig. 31. Gleason Templet-con

principle is shown in diagramma a spindle carried by a head whic the bed of the machine so that t be brought to point O by means equipment of the machine.

Three templets are used, moun of the bed, on the further side in templets is for "stocking" or roug the tool to cut a straight gash in the stock. After each tooth spa the templet holder is revolved t into position, and a tool is set i

the same relation to the shape of the tooth desired as the cam roll does to the templet. The head is again fed in by swinging it around its vertical axis, during which movement the roll runs up on the stationary templet, swinging the tool about its horizontal axis in such a way as to duplicate the desired form on the tooth of the gear. One side of each tooth being thus shaped entirely around, the holder is again revolved to bring the third templet into position. This has a reverse form from the preceding one adapted to cutting the other side of the tooth. A tool with a cutting point facing the other way being inserted in the holder, each tooth of the gear has its second side formed automatically, as before, completing the gear. The swinging

Fig. 32. The Bilgram Bevel Gear Generating Machine

movement for feeding the tool and the indexing of the work are taken care of by the mechanism of the machine without attention on the part of the operator.

Bevel Gear Generating Machines

The mechanism illustrated in outline in Fig. 29 is one that has been employed in a number of interesting and ingenious machines. The first application of this principle was made by Mr. Hugo Bilgram of Philadelphia, Pa. This form of machine in the hand-operated style has been used for many years. An example of a more recently developed automatic machine of the same type is shown in Fig. 32. The movements operate on the same principle as in Fig. 29, though in a modified form. Instead of rotating the crown gear and master gear

together, the imaginary cro
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is rotated about the axis
gear on the latter and roll
Instead of using crown an
pitch cone of the master g
representing the pitch sur
are prevented from slippi
stretched so as to make the
consists in extending the w
mounting the blank on th
being also on the lower s
shown in the diagram. A
32. As explained, a tool

Fig. 33. Glea

the side of a rack tooth,
crank, adjustable to vary
Whitworth quick-return
effected by swinging the
supports are hung, about t

As stated, the machine i
and places a previously-ga
the tool in operation. Th
attention, complete one si
then readjusted and the to
be finished in the same a
operate on the principle
going to the next. It foll

stroke of the tool, the rolling action being progressive with the indexing so as to finish all the teeth at once.

The Gleason generating machine is shown in Fig. 33. It differs from the previous machine in employing two tools, one on each side of the tooth. The construction is identical with the mechanism in Fig. 29, in having the axes of the tool-slides and of the blank fixed in relation to each other during the operation, the tool-holders and the blank rocking about their axes to give the rolling movement for cutting. The rocking is effected by means of segments of an actual crown gear and master gear. The segment of the crown gear is permanently attached to the face of the rear of the cutter slide frame, while the segment of the master gear (of which there are several furnished with the machine, the one used being chosen to agree with the angle of the gear to be cut) is clamped to the semi-circular arm pivoted at the outer end of the machine at one side, and fastened to the work spindle sleeve on the other. This arm is rocked by a cam mechanism and slotted link on the side opposite that shown in the illustration.

The cycle of operations is as follows: The machine being adjusted properly in its preliminary position, the tool-slide and the head on which it is mounted are swung back about the vertical axis so that the tools clear the work. The blank being set in the proper position, a cam movement swings the cutter slide head inward until the reciprocating tools reach the proper depth. The cam movement first mentioned now rocks upward the semi-circular arm extending around the front of the machine, rolling the blank and (through the segmental crown and master gears) the slide, until the tools have been rolled out of contact in one direction, partially forming the teeth as they do so. The arm is then rolled back to the central position and along downward to the lower position, until the tools are rolled out of contact with the tooth in this direction, completing the forming of the proper shape as they do so. The cam then rocks the arm back to the central position, where the cutter-slide head is swung back to clear the tooth, and the work is indexed, after which this cycle of operations is continued for the next tooth. It will be seen that by starting from the central position, going to each extreme and returning, all parts of each tooth are passed over twice, giving a roughing and a finishing chip. The machine is entirely automatic.

CHAPTER VII

CUTTING THE TEETH OF BEVEL GEARS

Special directions for operating are furnished by the makers of molding-generating and templet planing machines. As these directions are usually adequate, and apply only to the particular machines for which they are given, this chapter will be confined to giving instructions for cutting teeth by the formed tool method only, as performed on standard machine tools.

The Practicability of the Formed Tool Process

The first piece of instruction to be given in cutting bevel gears with a milling cutter is—don't do it. There are exceptions a-plenty to this rule, of course. For instance, gears too small to be cut on any commercial planing machine may be milled with a formed cutter; in general, it is not considered advisable to plane gears having teeth finer than 12 to 16 diametral pitch. It is allowable, also, to mill gears of coarser pitch which are to run at slow speeds or which are to be used only occasionally—such, for instance, as the bevel gears used for driving the elevating screws of a planer cross-rail, or those used in connection with any hand-operated mechanism. It is impracticable under ordinary conditions to mill teeth of bevel gears having teeth coarser than 3 diametral pitch, no matter what the service for which they are to be used.

Cutting Bevel Gears in the Milling Machine

The first requirement for setting up the milling machine to cut bevel gears is a true-running blank, with accurate angles and diameters. If such a blank cannot be found in the lot of gears to be cut, it will be necessary to turn up a dummy out of wood or other easily worked material. Otherwise the workman is inviting trouble, whatever his method of setting up.

Fig. 34 shows the machine set up for cutting a bevel gear, and Fig. 35 shows in diagram form the relative positions of the cutter and the work. The spindle of the dividing head is set at the cutting angle, as shown, and the cutter (which has been centered with the axis of the work-spindle) is sunk into the work to the whole depth W, as given by the working drawing.

The Brown & Sharpe Mfg. Co. recommends that for shaping with a formed cutter, the cutting angle be determined by subtracting the *addendum* angle from the pitch cone angle, instead of subtracting the dedendum angle as in Rule 15. In other words, the clearance at the bottom of the tooth is made uniform, as shown in Fig. 37, instead of tapering toward the vertex. This gives a somewhat closer approximation to the desired shape.

The centering may be done by mounting a true hardened center in the taper hole of the spindle, and lining up its point with the mark which will be found inscribed either on the top or on the back face of the tooth of the commercial gear cutter. A more accurate method is described in MACHINERY's Shop Operation Sheet, No. 1. Setting the cutter to the whole depth W is effected by passing the work back and forth under the revolving cutter and slowly raising it until the teeth of the cutter just bite a piece of tissue paper laid over the edge of the blank. This must be done after centering. The dial on the elevating screw shaft is set at zero in this position, and then the knee is raised an amount equal to the whole depth of the tooth, reading the dial from zero. This is evidently not exactly right, since the measurement should be taken in the direction of the back edge of the tooth, which inclines from the perpendicular an amount equal to the dedendum angle, as

Fig. 34. Milling Machine Set Up for Cutting a Bevel Gear

shown in Fig. 35. In practice, the slight difference in the value for the whole depth thus obtained is negligible.

Having thus mounted the work at the proper angle and having thus centered the cutter and set it to depth, two tooth spaces should next be cut, with the indexing set by the tables furnished with the dividing head to give the number of teeth required for the gear. Cutting these two spaces leaves a tooth between on which trial cuts are to be made until the desired setting is obtained. The relative positions of the cutter and the work and the shape of the cuts thus produced are shown in the upper part of Fig. 35. It will be seen at once that this does not cut the proper shape of tooth. As explained in the first paragraph in Chapter VI, all the elements of the bevel gear tooth vanish at O, the vertex of the pitch cone—that is to say, the outer corners of the tooth space should converge at O instead of at A, and the sides of the tooth spaces at the bottom, instead of having the parallel width

given them by the formed cutte
next problem is that of so re-sett
teeth as nearly as possible like t
ments converge at *O*.

**Offsetting and Rolling
the Sha**

There are a number of ways o
bevel gear teeth. Of these we hav

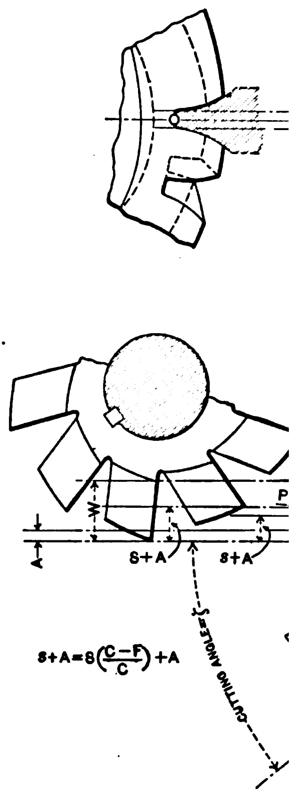
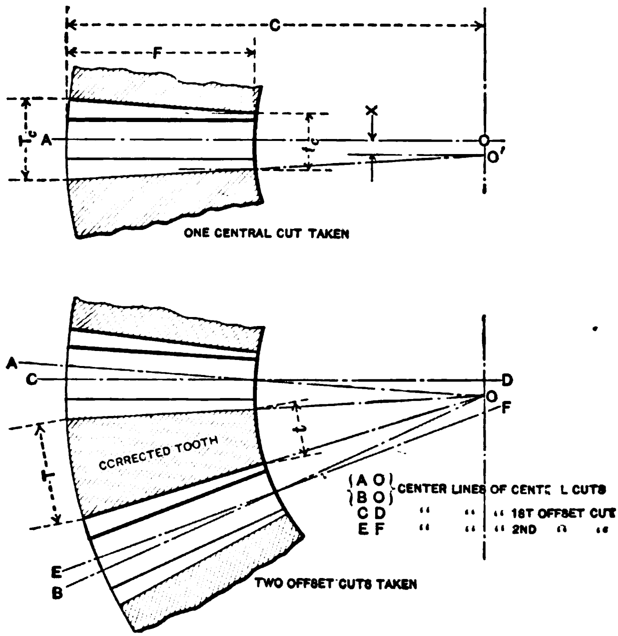


**Fig. 35. Relative Positions o
when takir**

in which the sides of the tooth
toward the vertex of the pitch c
show, of course, the proper thickn
by the gear tooth caliper at eith
method of approximation produces

are somewhat too wide at the bottom and too narrow at the top, or, in other words, the teeth themselves at the small end are too narrow at the bottom and too wide at the top. To make good running gears they must be filed afterward by hand, as described later. When so filed they are better than milled gears cut by other methods of approximation which omit the hand filing.

In the upper part of Fig. 36 is shown a section of the gear in Fig. 35, taken along the pitch cone at *PO*. It will be seen that the teeth at the pitch line converge, but meet at a point considerably beyond the vertex *O*. What we have to do is to move the cutter off the center, so that it will cut a groove, one side of which would pass through *O* if extended that far. The amount by which the cutter is set off the center



Machinery, N.Y.

Fig. 36. Section on Pitch Cone Surface PO of Fig. 35, showing Central and Offset Cuts

is known as the "set-over." We may take, for instance, for trial a set-over equal to 5 or 6 per cent of the thickness of the tooth at the large end. Move the face of the trial tooth away from the cutter by the amount of this trial set-over, having first, of course, run the cutter back out of the tooth space. Now rotate the dividing head spindle to bring this tooth face back to the cutter again, stopping it where the cutter will about match with the inner end of the space previously cut. Take a cut through in this position.

Next index the work to bring the cutter into the second tooth space and move the blank over to a position the other side of the central position by an amount equal to the same set-over, thus moving the

opposite face of the trial tooth away from the cutter. Rotate the dividing head spindle again to bring this face toward the cutter until the latter matches the central space already cut at the inner end of the teeth. Take the cut through in this position.

Now with vernier gear tooth calipers or with fixed gages machined to the proper dimensions measure the thickness of the tooth at the pitch line at both large and small ends (the values for the addendum and the thickness of the pitch line at both ends of the tooth are given by Rules 5, 8, 10 and 11). If the thickness is too great at both the large and the small ends, rotate the tooth against the cutter and take another cut until the proper thickness at either the large or small end has been obtained. If the thickness comes right at both ends the amount of set-over is correct. If it is right at the large end and too thick at the small end, the set-over is too much. If it is right at the small end and too thick at the large end the set-over is not enough. The recommended trial set-over (5 or 6 per cent of thickness of the tooth at the pitch line at the large end) will probably not be enough, so two or three cuts will have to be taken on each side of the trial tooth, as described, before the proper amount is found.

Having found the proper set-over, the cross-feed screw is set to that amount and the cut is taken clear around the gear. Then the cross-feed screw is set to give the same amount of set-over the other side of the center line and the work is rotated until the cutter matches the tooth spaces already cut at the small end and is run through the work. The tooth will generally be found too thick, so the work spindle is rotated still more until the tooth is of the proper thickness, when the gear is again cut clear around on this second cut.

The number of holes it was necessary to move the index pin on the dividing plate circle between the first and the second cuts to get the proper thickness of tooth, should be recorded. On succeeding gears it will thus only be necessary to take a first cut clear around with the work set over by the required amount on one side of the center line, and then a second cut around with the work set over on the other side of the center line, rotating the index crank the number of holes necessary to give the proper thickness of tooth between the cuts.

It will be noted that the shifting of the blank by the index crank is only used for bringing the thickness of tooth to the proper dimension. In some cases, particularly in gears of fine pitch and large diameter, this adjustment will not be fine enough—that is to say, one hole in the index circle will give too thick a tooth and the next one too thin a tooth. To subdivide the space between the holes, most dividing heads have a fine adjustment for rotating the worm independently of the crank. Every milling machine should be provided with such an adjustment.

In large gears it is best to take the central cuts shown in Fig. 35 clear around every blank before proceeding with the approximate cuts. This gives the effect of roughing and finishing cuts, and produces more accurate gears. The central cuts may be made in a separate operation with a roughing or stocking cutter if desired. It might also be men-

tioned that it is common practice to turn up a wooden blank for making the trial cuts shown in Figs. 35 and 36, to avoid the danger of spoiling the work by mistakes in the cut-and-try process.

Positive Determination of the Set-over*

This cut-and-try process, however, may be practically eliminated by calculating the set-over from the following table and formula:

TABLE FOR OBTAINING SET-OVER FOR CUTTING BEVEL GEARS

No. of Cutter	Ratio of Pitch Cone Radius to Width of Face $\left(\frac{C}{F}\right)$												
	$\frac{3}{1}$	$\frac{3\frac{1}{4}}{1}$	$\frac{3\frac{1}{2}}{1}$	$\frac{3\frac{3}{4}}{1}$	$\frac{4}{1}$	$\frac{4\frac{1}{4}}{1}$	$\frac{4\frac{1}{2}}{1}$	$\frac{4\frac{3}{4}}{1}$	$\frac{5}{1}$	$\frac{5\frac{1}{2}}{1}$	$\frac{6}{1}$	$\frac{7}{1}$	$\frac{8}{1}$
1	.254	.254	.255	.256	.257	.257	.257	.258	.258	.259	.260	.262	.264
2	.266	.268	.271	.272	.273	.274	.274	.275	.277	.279	.280	.283	.284
3	.266	.268	.271	.273	.275	.278	.280	.282	.283	.286	.287	.290	.292
4	.275	.280	.285	.287	.291	.293	.296	.298	.298	.302	.305	.308	.311
5	.280	.285	.290	.293	.295	.296	.298	.300	.302	.307	.309	.313	.315
6	.311	.318	.323	.328	.330	.334	.337	.340	.343	.348	.352	.356	.362
7	.289	.298	.308	.316	.324	.329	.334	.338	.343	.350	.360	.370	.376
8	.275	.286	.296	.309	.319	.331	.338	.344	.352	.361	.368	.380	.386

For obtaining the set-over by the above table, use this formula:

$$\text{Set-over} = \frac{T_c}{2} - \frac{\text{factor from table}}{P}$$

P = diametral pitch of gear to be cut,

T_c = thickness of cutter used, measured at pitch line.

Given as a rule this would read: *Find the factor in the table corresponding to the number of the cutter used and to the ratio of pitch cone radius to width of face; divide this factor by the diametral pitch, and subtract the result from half of the thickness of the cutter at the pitch line.*

As an illustration of the use of this table in obtaining the set-over we will take the following example: A bevel gear of 24 teeth, 6 pitch, 30 degrees pitch cone angle and 1¼ face. These dimensions, by the rules given in Chapter I, call for a No. 4 cutter and a pitch cone radius of 4 inches.

In order to get our factor from the table, we have to know the ratio of pitch cone radius to width of face. This ratio is $\frac{4}{1.25} = \frac{3.2}{1}$ or about $\frac{3\frac{1}{4}}{1}$. The factor in the table for this ratio with a No. 4 cutter is 0.280. We next measure the cutter at the proper depth of $S + A$ for 6 pitch, which is found in the column marked "depth of space below pitch line" in a regular table of tooth parts, or by dividing 1.157 by the diametral pitch. This gives $S + A = 0.1928$ inch. We find, for in-

* MACHINERY, December, 1909.

stance, that the thickness of the cutter at this depth is 0.1745 inch. The dimension will vary with different cutters, and will vary in the same cutter as it is ground away, since formed bevel gear cutters are commonly provided with side relief. Substituting these values in the formula, we have

$$\text{Set-over} = \frac{0.1745}{2} - \frac{0.280}{6} = 0.0406''.$$

which is the required dimension.

With reference to the use of the above table and formula, the Brown & Sharpe Mfg. Co., after trial in its gear-cutting department, says: "We feel fairly confident it is within working limits of being satisfactory." While this sounds encouraging, it will evidently be wise to be

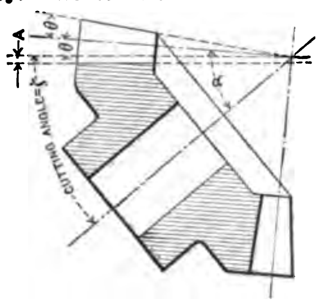
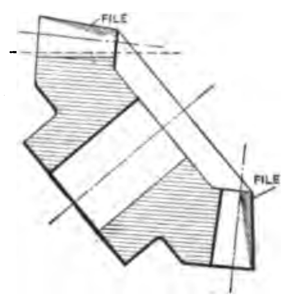


Fig. 37. Cutting Angle and Parallel Clearance Recommended by Brown& Sharpe for Cutting with Formed Cutter



Machinery, N.Y.

Fig. 38. The Surfaces to be Filed, in Fitting Bevel Gears Cut with a Formed Cutter

sure we are right before going ahead. So the trial tooth should be measured, the same as when the cut-and-try process is used.

Use of the Formula for Other Methods of Correction

It is customary also among workmen expert in cutting bevel gears with formed cutters, to cut loose from rules and formulas for the selection of the cutters, and depend on their experience to get shapes which require somewhat less filing than would otherwise be necessary. Whenever this "cutting loose" requires, as it sometimes does, the use of a cutter of finer pitch than that of the teeth of the bevel gear at the large end, the values given in the table are inapplicable. The following formula may then be used:

$$\text{Set-over} = \frac{T_c}{2} - \frac{T_c - t_c}{2} \times \frac{C}{F}$$

in which t_c is the thickness of the cutter measured at a depth $s + A$, obtained as shown in Fig. 35. This has been tried on several widely varying cases with good results. It requires, it will be seen, two measurements of the cutter in place of the single one required when the regular pitch of cutter is used.

Filing the Teeth

The method of cutting bevel gears just described requires the filing of the points of the teeth at the small end. This can be done "by the

eye" very skillfully when the workman is used to it. The operation consists in filing off a triangular area extending from the point of the tooth at the large end to the point at the small end, thence down to the pitch line at the small end and back diagonally to the point at the large end again. This is shown in Fig. 38 by the shaded outline. Enough is taken off at the small end of the tooth so that the edges of the teeth at the top appear to converge at vertex *O*.

The bevel gears may be tested for the accuracy of the cutting and filing by mounting them in place in the machine and revolving them at high speed, or by mounting them in a testing machine made for the purpose. The marks of wear produced by running them together under pressure, with the back faces flush with each other, should extend the whole length of the tooth at the pitch line. If it does not, the amount of set-over allowed in cutting them was at fault, being too little if they bear heavily at the large ends, and too much if they bear heavily at the small ends. The bearing area should also be fairly evenly distributed over the sides of the teeth above the pitch line, from the large to the small end. If it is not, the filing is at fault. The marks of wear will not extend far below the pitch line in a pinion of few teeth.

It is possible to get along without filing by decreasing the amount of set-over so as to make the teeth too thin at the pitch line at the small end, when they are of the right thickness at the large end. This does not give quite as good running gears, however, as when the method just described is followed.

Cutting Bevel Gears on the Automatic Gear-cutting Machine

The directions for cutting bevel gears on the milling machine apply in modified form to the automatic gear-cutting machine as well. The set-over is determined in the same way, but instead of moving the work off center, the cutter spindle is adjusted axially by means provided for that purpose. Some machines are provided with dials for reading this movement. The cutter is first centered as in the milling machine, and then shifted—first to the right, and then to the left of this central position.

The rotating of the work to obtain the proper thickness of tooth is effected by unclutching the indexing worm from its shaft (means usually being provided for this purpose) and rotating the worm until the gear is brought to proper position. Otherwise the operations are the same as for the milling machine.